MODERN AVIATION ENGINES

VOLUME ONE

NOTE

Volume One contains Chapters One to Twenty-seven inclusive—Pages 1 to 976.

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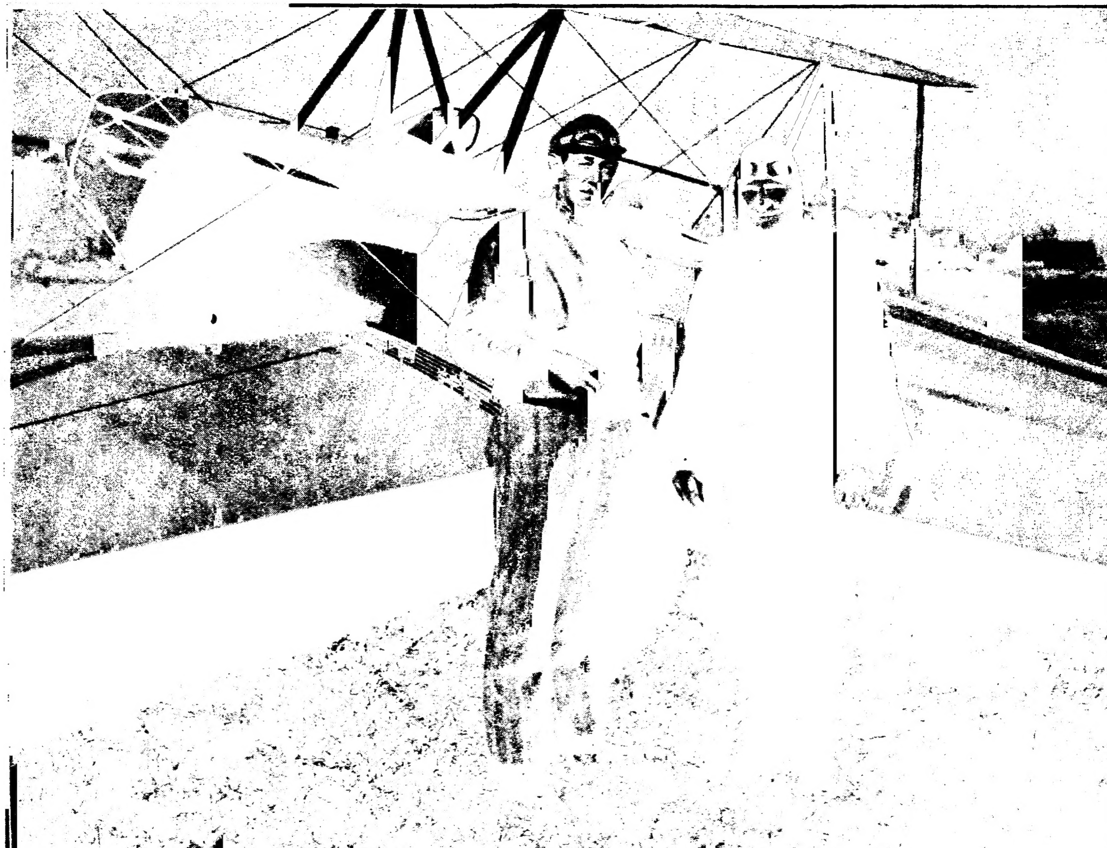

Hon. John H. Trumbull, Governor of the State of Connecticut Who is an Expert Airplane Pilot, is Shown in Flying Costume at the Right With His Teacher, Lieut. Harry D. ("Safety First") Copland, A.C. C.N.G., Who Has an Enviable Record as a Flight Instructor.

MODERN AVIATION ENGINES

DESIGN—CONSTRUCTION—OPERATION AND REPAIR

A COMPLETE, PRACTICAL TREATISE OUTLINING CLEARLY THE ELEMENTS OF INTERNAL COMBUSTION ENGINEERING WITH SPECIAL REFERENCE TO THE DESIGN, CONSTRUCTION, OPERATION AND REPAIR OF AIRPLANE POWER-PLANTS; ALSO THE AUXILIARY ENGINE SYSTEMS, SUCH AS LUBRICATION, CARBURETION, IGNITION AND COOLING

It Includes Complete Instructions for Engine Repairing and Systematic Location of Troubles, Tool Equipment and Use of Tools, also Outlines the Latest Mechanical Processes

IN TWO VOLUMES

BY

MAJOR VICTOR W. PAGÉ, U. S. AIR CORPS RESERVE

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Author of Modern Aircraft, Everybody's Aviation Guide, Etc.

Describes Many Typical American and European Engines and Their Installation. Contains Valuable Instructions for all Aviation Students, Pilots, Mechanicians, Flying Field Engineering Officers and All Interested in the Design, Construction and Upkeep of Airplane Powerplants.

VOLUME ONE

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THIS TREATISE IS RESPECTFULLY

DEDICATED TO

HON. JOHN H. TRUMBULL

GOVERNOR OF THE STATE OF CONNECTICUT

WHO PREACHES AVIATION AND

PRACTICES WHAT HE PREACHES

IN APPRECIATION OF THE EXAMPLE SET BY HIM

FOR THE COMING GENERATION TO EMULATE AND FOR

HIS WORK IN PROMOTING COMMERCIAL AVIATION

PREFACE

In presenting this treatise on "Modern Aviation Engines," the writer realizes that the rapidly developing art makes it difficult to outline all latest forms or describe all current engineering practice. This exposition has been prepared primarily for instruction purposes and is adapted for students who wish to become aviators or aviation mechanicians, and for mechanics in other lines who wish to enter the aviation industry as experienced aviation engine maintenance and repair men. Every effort has been made to have the engineering information accurate, but owing to the diversity of authorities consulted and use of data translated from foreign language periodicals, it is expected that some errors will be present. The writer wishes to acknowledge his indebtedness to many firms for photographs and helpful descriptive matter and endeavor has been made in every case to give credit to the firm furnishing such data, also to experts in various lines that have been quoted in this treatise. Special attention has been paid to instructions on tool equipment, use of tools, trouble "shooting" and engine repairs, as it is on these points that the average aviation student is weakest. Only such theoretical consideration of thermodynamics as was deemed absolutely necessary to secure a proper understanding of engine action (after consulting several experienced instructors) is included, the writer's efforts having been confined to the preparation of a practical series of instructions that would be of the greatest value to those who need a diversified knowledge of internal-combustion engine construction, operation and repair, and who must acquire it quickly. The engines described and illustrated are all practical forms that have been fitted to airplanes capable of making extended flights and may be considered fairly representative of the present state of the art.

Considerable space is devoted to the leading war-time engines in both the water- and air-cooled forms because some of these are still in use and also because these are the types from which our present day perfected engines have been developed and a review of their characteristics should be of value in showing the reader what has been done in the past, so he can better understand the possibilities of the future. As aviation and the increasing use of aircraft has practically eliminated national boundaries, this book has been made international in scope and many practical and successful European engines have been illustrated and described along with our own American product.

VICTOR W. PAGÉ.

March, 1929

ACKNOWLEDGMENT

One of the most important branches of aeronautical engineering is that dealing with powerplant design, construction, installation and repair and aeronautical engineers may be divided into two main groups, "plane" men and "engine" men. The division of the engine men is in three main classes; designers, builders or production men and field men who are concerned with installation, maintenance and repair. Specialists in any of the subdivisions find that it takes all their time to become familiar with the many phases of the subject they are interested in. While the author has had a broader experience than many of the specialists, it is only because he has been identified with aviation since its inception and because of particularly fortunate circumstances while serving on the Staff of the Chief of the Air Corps during the World War, which offered unexcelled opportunities to obtain experience on a larger scale than normal peace time activities permitted.

Regardless of this experience, the author has found it desirable and even necessary to consult other authorities and specialists in order to check up on his own opinions and experience and every effort has been made in this treatise to present both sides of every controversial subject. The reader may select the line of reasoning that best applies to the case under consideration and no matter what he finally accepts, he will find ample authority as a basis for his line of thought. In preparing this work the author has made references to the authority responsible for the opinions or information presented and in every case due acknowledgment is made in the text to the expert quoted, when the opinions are not those of the author.

There are many sources of aeronautical data at the present time besides the manufacturers of airplanes, engines and auxiliary apparatus. Government documents and publications of the Engineering Division, U. S. Army Air Corps, with laboratory facilities at Wilbur Wright Field, Dayton, Ohio; and also those of the National Advisory Committee for Aeronautics, Washington, D. C.; have been consulted freely and brief excerps and abstracts from these public documents have been used to bring out points in the text that were considered in greater detail in reports of experts and specialists. The United States Bureau of Standards, and the United States Department of Commerce, Washington, D. C., have also published much valuable data in the form of reports issued in co-operation with the Government agencies previously mentioned.

The membership of the Society of Automotive Engineers, Inc., includes many aeronautical experts and specialists and much valuable data has been published in the *S. A. E. Journal* on aviation and kindred subjects. The publications *Aviation* and *Aero Digest* of New York City were also of great value and references to editorial opinions and descriptions of aircraft engines, have also been included to justify and support some of the opinions of the author. Such leaders in the industry as the Goodyear-Zeppelin Co.,

Akron, Ohio; the Curtiss Aeroplane and Motor Company, Inc., of Garden City, New York; Packard Motor Car Co., Detroit, Mich.; the Wright Aeronautical Corporation, Paterson, New Jersey; Pratt & Whitney Aeronautical Corporation, Hartford, Conn.; as well as numerous other firms whose products are described in the text, furnished valuable illustrative and descriptive data. The Bureau of Aeronautics, U. S. Navy and the Information Section, U. S. Army Air Corps, also furnished material pertinent to service planes, airships, and engines. A number of early engines were described in Angle's *Airplane Engine Encyclopedia* which was also referred to in preparing this volume.

The writer desires to acknowledge the valuble assistance obtained from the sources mentioned as they have greatly supplemented the material in the original aviation engine instruction papers prepared for students and Army mechanics during the late War and the author's experience in aviation since its inception over two decades ago that forms the ground work for this treatise. The character of co-operation obtained cannot fail to promote knowledge of aviation and proper public appreciation of its great possibilities.

The public spirit and enthusiastic co-operation of the publisher of this treatise in going to an unusual expense in financing extended research work of the author and for the numerous excellent special illustrations that accompany the text, in order to help the cause of aviation and also in giving the writer *carte blanche* in the preparation of an unusually complete work without allowing purely business reasons to limit the size and scope, is also worthy of comment and appreciative acknowledgment.

VICTOR W. PAGÉ.

March, 1929

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HISTORICAL INTRODUCTION

THE FIRST ENGINES TO FLY

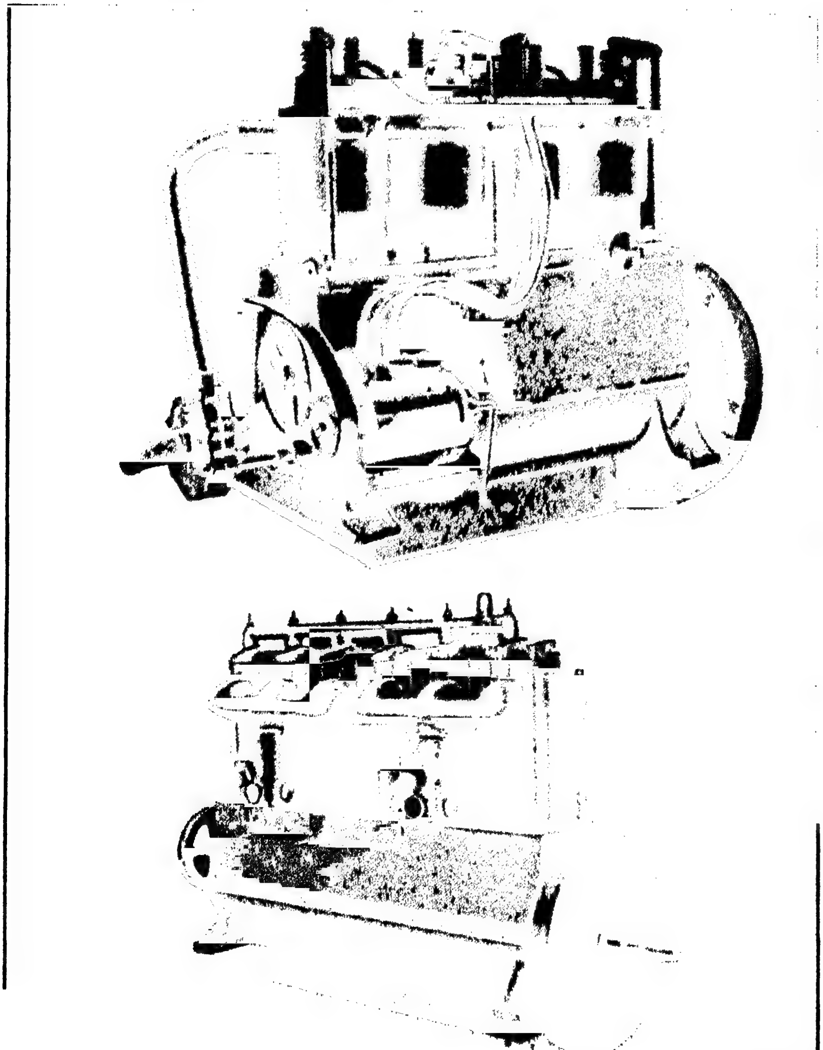
As the writer is dictating these words, the Aeronautical Industry is celebrating the twenty-fifth anniversary of flight and while the public is chiefly concerned with the airplane and other forms of air-craft, but little prominence is given in the public prints to the most important thing that has made mechanical flight possible, i.e., the highly refined internal-combustion engine.

The first engine to furnish power enough to cause an airplane to leave the ground and make a landing under control of the pilot was undoubtedly the Wright airplane engine, an early form, which is illustrated herewith. Various forms of electric motors, steam engines and even light gasoline engines of the motorcycle type had moved gas bags through the air prior to the development of the airplane but in these early airships, the powerplant was called upon only to move the gas bag and not to furnish power enough for sustentation as well.

The Wright Brothers found that there were no light engines of sufficient power available in 1902 to convert their motorless glider to a power driven machine. Motorcycle engines, which were sufficiently light, were not powerful enough and automobile type engines of adequate power were much too heavy, the minimum weights being from twelve to fifteen pounds per horsepower. The Wright brothers were forced to work out their own engine, which was a very creditable design and a big improvement over existing automotive powerplants of the period as far as weight-power ratio was concerned.

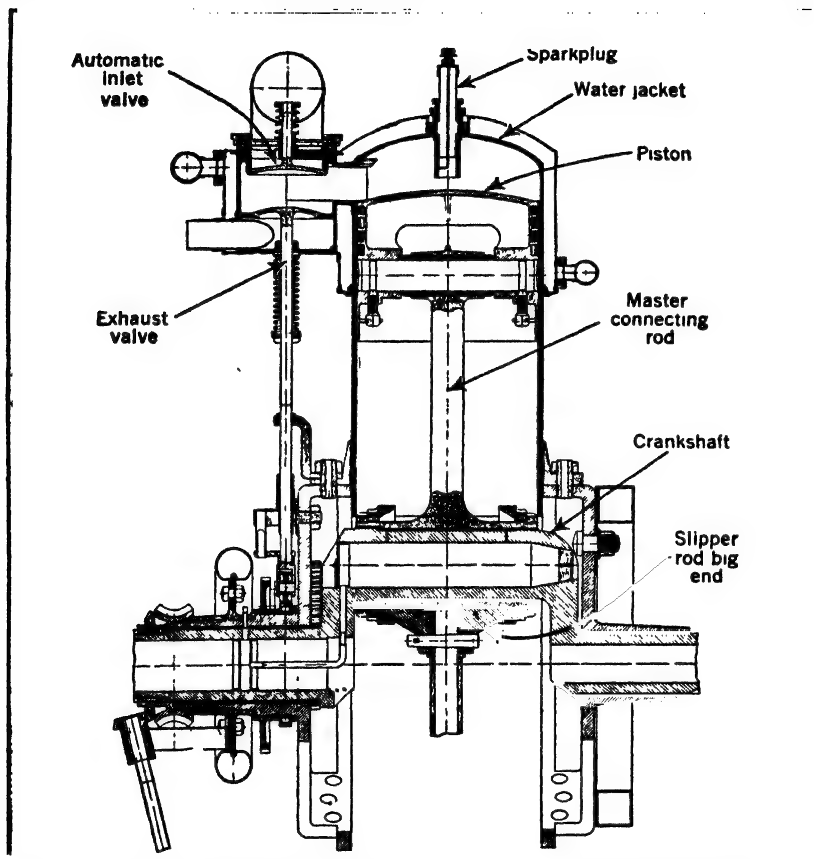
The first Wright engine was of the water-cooled vertical cylinder form, having four cylinders with a total displacement of approximately 240 cubic inches. The bore was 4⅜ inches and the stroke four inches. It developed 30 to 35 horsepower at 1,200 r.p.m. and weighed 180 pounds, giving a power loading of six pounds per horsepower. Cast-iron cylinders with applied sheet aluminum water jackets were used and the valves were placed in the cylinder head, the exhaust being mechanically actuated while the inlet was an automatic type. The crankcase was of aluminum alloy with an oil sump. Carburetion was by measured fuel injection into the manifold. Ignition was furnished by a Mea high-tension magneto. The crankshaft was machined from a solid billet of steel. This engine, which appears crude in the light of our present knowledge, was a real and basic contribution to the infant science or art of aerial navigation. The great saving in weight made possible by this pioneer design resulted in the success of the flying demonstrations made during 1903 which focussed the eyes of the world on the navigation of the air by heavier-than-air machines. Further refinements were made and six-cylinder in-line and eight-cylinder Vee types were evolved by the Wright brothers that had a still more favorable weight-power ratio than the original, getting down to a weight of four pounds per horsepower.

The first radial engine to fly a heavier-than-air machine was that used by Professor Langley in his flying experiments. This engine, designed by the late Charles M. Manly and built by its inventor was years ahead of its time. From the sectional view presented, the reader will pick out many features that are found in modern radial engines, though in refined forms. The Langley "Aerodrome" did not make a long flight and the Wright airplane did and for that reason, there was little development of radial engines except by Adams when he was connected with the Farwell interests, and most of the successful early flights were made with automobile type engines refined in design to secure light weight.



The Engine Shown at the Top is the Wright Brothers' Four-Cylinder Water-Cooled Type That Was the First Engine to Fly an Airplane. The Improved Six-Cylinder Design is Shown Below It.

The Manly engine was a five-cylinder water-cooled fixed radial form with a total displacement of 540 cubic inches. The cylinders were five-inch bore and five and one-half-inch stroke and it developed 52.4 horsepower with a weight of but 150 pounds, giving a power-weight loading of 2.86 pounds per horsepower. Aeronautical engineers should have realized the great weight saving made possible by the radial disposition of the cylinders in the Manly engine but this interesting pioneer form had little

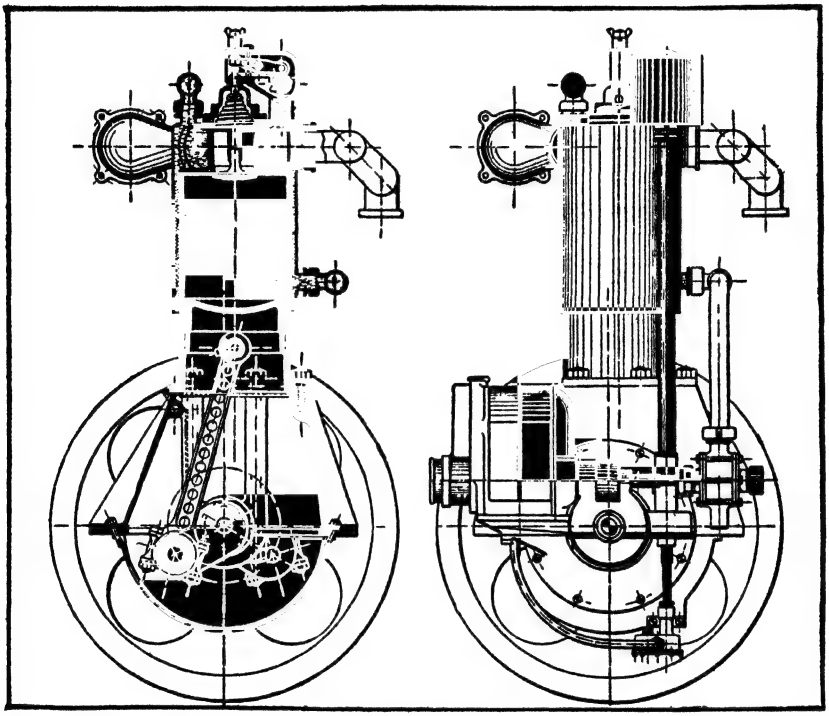


The Manly Engine Built for Professor Langley's "Aerodrome" was the Pioneer Form of Radial Engine.

influence on aero engine design for a number of years. The real excellence of the Manly engine did not receive the recognition it deserved, but its inventor had the satisfaction of living long enough to see radial engines displacing other types in commercial and military airplanes.

The cylinders were of the "L" head type, built of sheet steel with pressed in iron liners and sheet steel jackets were brazed on, the autogenous welding process not having been developed except as a laboratory novelty at that time, so joints now made by oxy-acetylene or torch welding were obtained either by silver soldering or brazing. The intake valves were automatic

for the sake of simplicity as were those of the early Wright engine, the early Curtiss and the Anzani. It was in the arrangement of the connecting-rod assembly that great ingenuity was shown and the system evolved by Manly was later used in modified form on many successful engines. The master rod was a solid section but the link rods were tubular section. The link rod ends were provided with bearing "feet" that pressed against the master rod and that were retained by adjustable clamping rings having



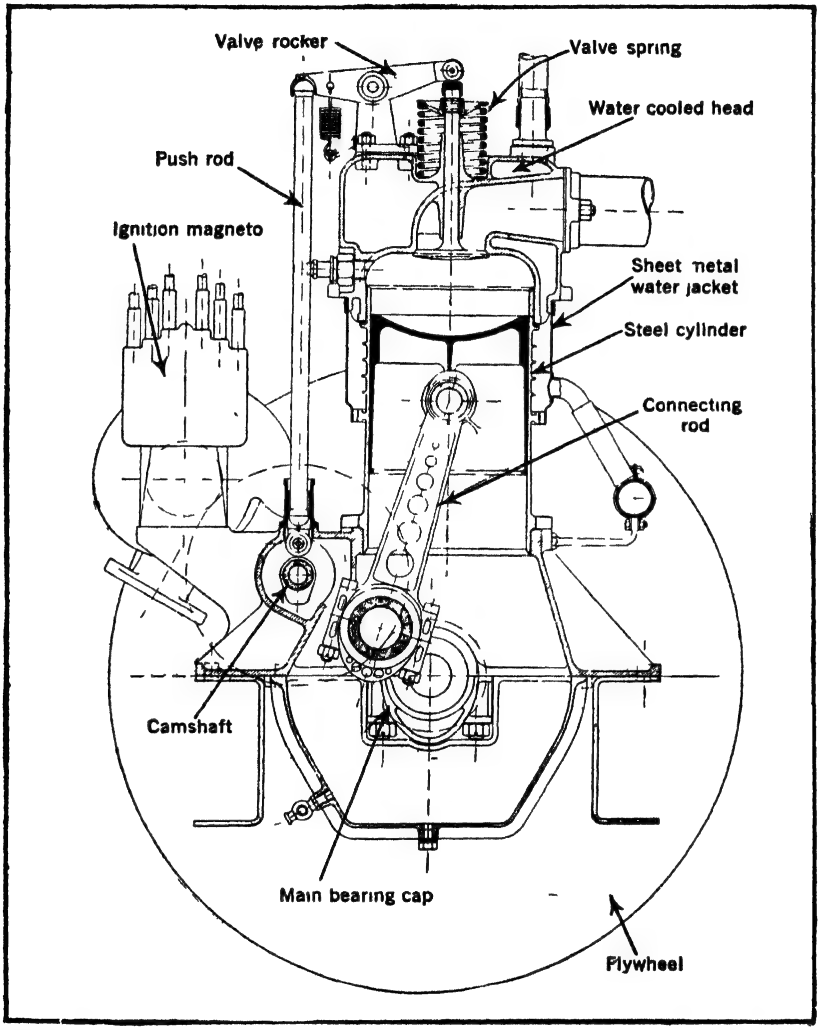
The Early Green Engine Which was One of the First to Make Successful Flights in England is Shown Above in Transverse Section and End View.

female tapers to act against the male taper on the link rod big ends. The extreme lightness of the various parts, the boring out of all solid sections such as the crankshaft and link rods, the light sections of the cylinder and water jacket were all features that showed the exceptional engineering ability which the inventor possessed.

Glenn Curtiss, another pioneer in aero engine construction, combined four motorcycle type air-cooled cylinders on one base and produced a successful four-in-line engine. He afterward made an eight-cylinder Vee type because he realized the need for more power, but his engines did not fly airplanes until the Wright Brothers had shown the way to the rest of the experimenters.

In Europe, Levasseur built a light engine known as the Antoinette and Duthiel-Chalmers made a twin opposed with which Santos-Dumont made flights in a diminutive airplane of his own design. In England, the Green

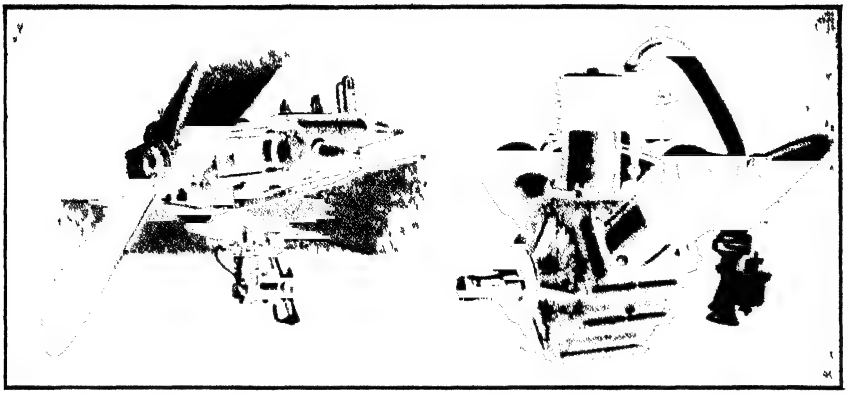
engine was an early water-cooled type patterned after the Wright. The man responsible for most of the early fixed radial engine development in Europe was Anzani, who had a line ranging from two to twenty cylinders available in a very short time. All of these engines were designed after the Wright Brothers had flown their airplane, so the four-cylinder Wright



The Transverse Section of the German A. E. G. Engine Shows the Influence of the Wright Brothers' Pioneer Design on Early Flight Motor Construction.

engine will remain in history as the pioneer form and that which made the first controlled flight; because an unlucky break of a launching mechanism, which resulted in a ducking in the Potomac river for Manly and the Langley Aerodrome, prevented the Manly radial engine from obtaining the coveted first place. Many authorities, without seeking to diminish the

prestige gained by the Wright Brothers, believe that only an unfortunate accident prevented Manly from being the first man to fly and the Langley "Aerodrome," a twin tandem monoplane, from being the first heavier-than-air machine to carry an aviator under power. Langley's model, "Aerodrome," propelled by a steam engine had made a flight of three-quarters of a mile previous to the flight of the Wright Brothers' machine, but it was not a full scale machine nor could it carry a pilot.



Early French Aviation Engines That Made Flights. The Dutheil-Chalmers Flat Twin Shown at Left Was Used by Santos-Dumont, the Anzani Three Cylinder Fan Type Was Used by Bleriot.

An early Italian development was the Fiat air-cooled engine, made by the well known motor car building firm, which is still building aero engines but of the water-cooled form.

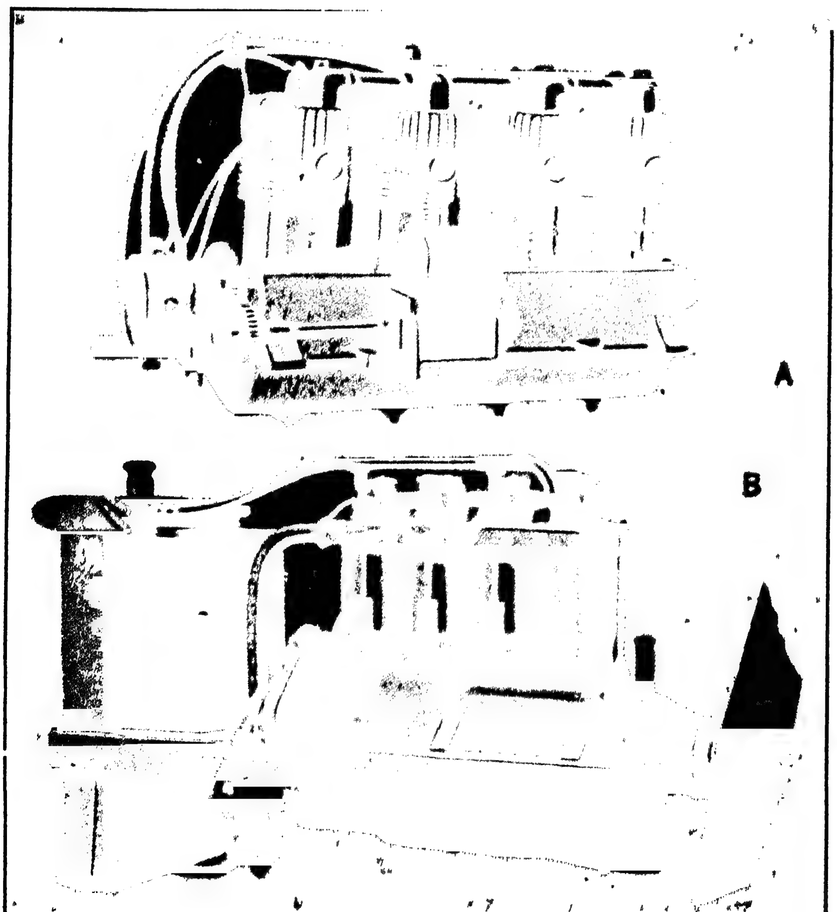
This engine was of the eight-cylinder Vee type and rated at 50 horsepower, weighing about 150 pounds. The cylinders were the conventional flange-cooled type, and had separate heads, held in place by through bolts going into the crankcase. Both inlet and exhaust valves were located so they opened directly into the cylinder and were of large diameter. The heads were cast iron and deeply ribbed to make for positive cooling. The exhaust was direct to the air by a short piece of pipe leading from the valve chamber in the head. The cylinders were disposed in the conventional manner, and the valves were operated through a single camshaft by overhead tappet levers and push rods.

As will be seen from the illustration the lower half of the motor was enclosed in a hemispherical shield, and a similar member, removed to make the details of the motor clearer, was used over the cylinders, this enclosing the entire motor in a cylindrical hood. As the propeller was driven direct from the crankshaft in the usual installations, the slipstream created by its turning would force air currents through this casing, and cool the cylinders. Where the motor was to be applied to driving propellers by chain or gears the cooling was by a fan placed at the front or back of the casing as was most convenient, driven by the engine in the usual manner.

Two separate magnetos were used for ignition, one for each set of four cylinders, but a single carburetor served to supply gas to all eight. Every

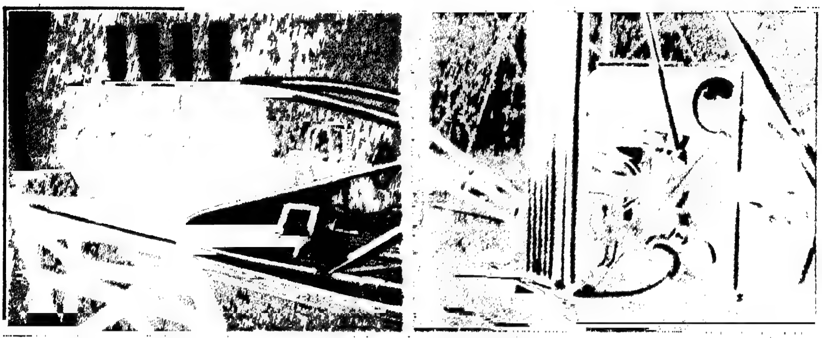
endeavor was made to lighten the construction, this being well shown by the skeleton form of the walking beams that operated the valves, the hollow camshaft and extremely light spoked construction of the valve and magneto gears. This engine gave excellent results in brake tests under full loads for periods of three and four hours, without signs of overheating.

Prominent among the early air-cooled motors that were distinctive in design were the creations of Robert Esnault-Pelterie, and while the general details did not differ radically from conventional four-cycle practice, the arrangement of the cylinders, placed around the engine base at various angles and in general appearance being suggestive of a huge fan was as shown in the photograph. These motors were manufactured in three principal forms, of five, seven and ten cylinders, ranging in power from 25 to 50 horsepower. The motor shown in the photographic reproduction is the



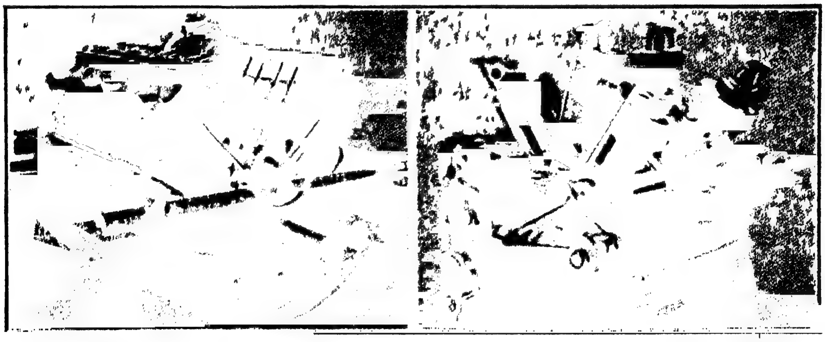
The Four-Cylinder Air-Cooled Engine at A was the Pioneer Curtiss Form and was Built with Motorcycle Type Cylinders. A Later Water-Cooled Form Which was the Parent of the Famous Curtiss OX Series is Shown at B.

ten-cylinder form, which is composed of two five-cylinder groups assembled on a common base. The ten-cylinder type, had many interesting features, combined with a novel arrangement of the cylinders, grouped in the same fan-like manner that was characteristic of these motors. The cylinders are arranged in pairs, one pair being vertical, two pairs at an angle and the other two nearly horizontal.



Two Early Airplane Engine Installations. At Left, the Antoinette Eight-Cylinder Vee Type. At Right, Showing How Wright Brothers' Engine was Placed Beside the Pilot.

Another of the early light-weight air-cooled engines worthy of comment was the eight-cylinder Farcot engine, the invention of a well known French designer. In this construction, lightness was obtained in two ways. First by making the cylinders air cooled by means of ordinary heat radiating flanges cast integral, and secondly by grouping the cylinders horizontally around the crankshaft. While this arrangement of components around a center engine base was not a new one it had not received the attention that its merit deserved at that time. The accompanying illustration shows the



The Early Fiat Airplane Engine Shown at the Left was an Eight-Cylinder Air-Cooled Vee Type. The R.E.P. Engine Shown at the Right was an Early Fan Type Air-Cooled Powerplant.

general arrangement of the eight cylinders about the crankcase and, as will be evident, four of the cylinders are necessarily in a different plane from the others. This construction is necessary that the connecting rods can be attached to the crank, which is really a double throw shaft as used on the

ordinary double opposed motor. Four cylinders act against the same crank-pin. The two cranks are set at 180 degrees, and alternate cylinders are set in different planes. As the crankshaft is set vertically, a horizontal shaft driven by bevel gears at a somewhat reduced speed, is located near the top of the crankshaft for the purpose of driving the propeller shaft. Ball bearings were used to take up the end thrust of the bevel gears, while the crankshaft revolved in three bearings.

Owing to the construction described, it was found possible to reduce the weight to about 2.2 pounds per horsepower, a figure not to be despised even in these days. This included the double ignition system, the four carburetors, the fan and the oiler. The eight-cylinder motor was made in three different sizes of 30, 50 and 100 horsepower respectively. The cylinder dimensions of the smaller of these was 3.14 inches by 3.5 inches, the total weight being with complete equipment, 84 pounds, and the power being developed at the normal speed of 1,800 revolutions per minute. The speed of rotation also corresponds to our modern engines even though the design illustrated is over twenty years old.

The Farcot Air-Cooled Engine Shown at Left was an Early Static Radial Form with Cylinders Horizontally Placed. Note Large Cooling Fan. The Renault Engine at the Right was Air-Cooled by Blower and Air Jackets.

The 50 horsepower engine was 4.13-inch bore by about 4¾-inch stroke, weighed 121 pounds, and developed its rated horsepower at about 1,500 revolutions per minute. The largest motor was 5.11-inch bore and 5.31-inch stroke and was said to have produced 110 horsepower at a speed of 1,200 revolutions per minute. The large fan, positively driven by direct attachment to the crankshaft, forces an unfailing draught of air against the cylinders, which makes it possible to operate the engine continuously without excessive overheating, but this fan should not be confused with the propeller furnishing traction as that member was of the usual two blade design and was driven by a separate shaft. This motor was designed to be installed in the fuselage of the airplane and was intended to drive the air screw at the front or rear by fastening a shaft to the taper end of the power take-off shown extending from the crankcase under the cooling fan.

Another example of the early light weight motor of French development is shown, this being an eight-cylinder Vee type engine rated at 50 horse-

power built by Renault that had individual cylinder units with heat radiating flanges cast integral. As will be evident the cylinders are of the conventional type, the heads being a separate construction held to the cylinder by through bolts attached to a special fitting, or cross casting, designed to strengthen the head, which was made very light. The aluminum casing which directed the air blast produced by the fan blade spoked rotor against the cylinders is clearly shown, as is the placing of the exhaust valves and pipes. The carburetor was attached to a distributing manifold and was of the regular float feed type, this component having been supported by a bracket attached to the casing. The inlet valves were located in the lower portion of the valve pocket, operated from the camshaft directly, while the exhaust valves were actuated by means of a walking beam and rod connection.

The bore and stroke were approximately 3½ and 4¾ inches, respectively. As was common with most Vee motors the valves were mechanically operated from a single camshaft, the exhaust valve was placed above the inlet. The motor was provided with a carburetor made of aluminum, to secure light construction. The ignition was by high-tension magneto. The cooling was produced by a fan or blower of large diameter which forced air through the chamber formed by the cylinders and their containing case of sheet metal. It will be evident that as the air could escape only after circulating around and over the flanges of the cylinders positive cooling was obtained. The motor developed over 55 horsepower at 1,800 revolutions per minute, which is 300 turns faster than its normal speed, but owing to the use of the air-cooling blower, it did not have as good weight-power ratio as those engines of the Vee or radial form that depended on the propeller slipstream for cooling the cylinders.

It will be apparent to the reader of the chapters to follow that great progress has been made in the detail refinement and construction of modern aviation engines, but these improvements can only be appreciated by comparing the present day forms with the pioneer engines illustrated in this historical review. Basic principles are the same, but proportioning and construction of parts and the better materials employed in new engines give greatly increased efficiency and reliability over the performance of the engines used by pioneer aviators.

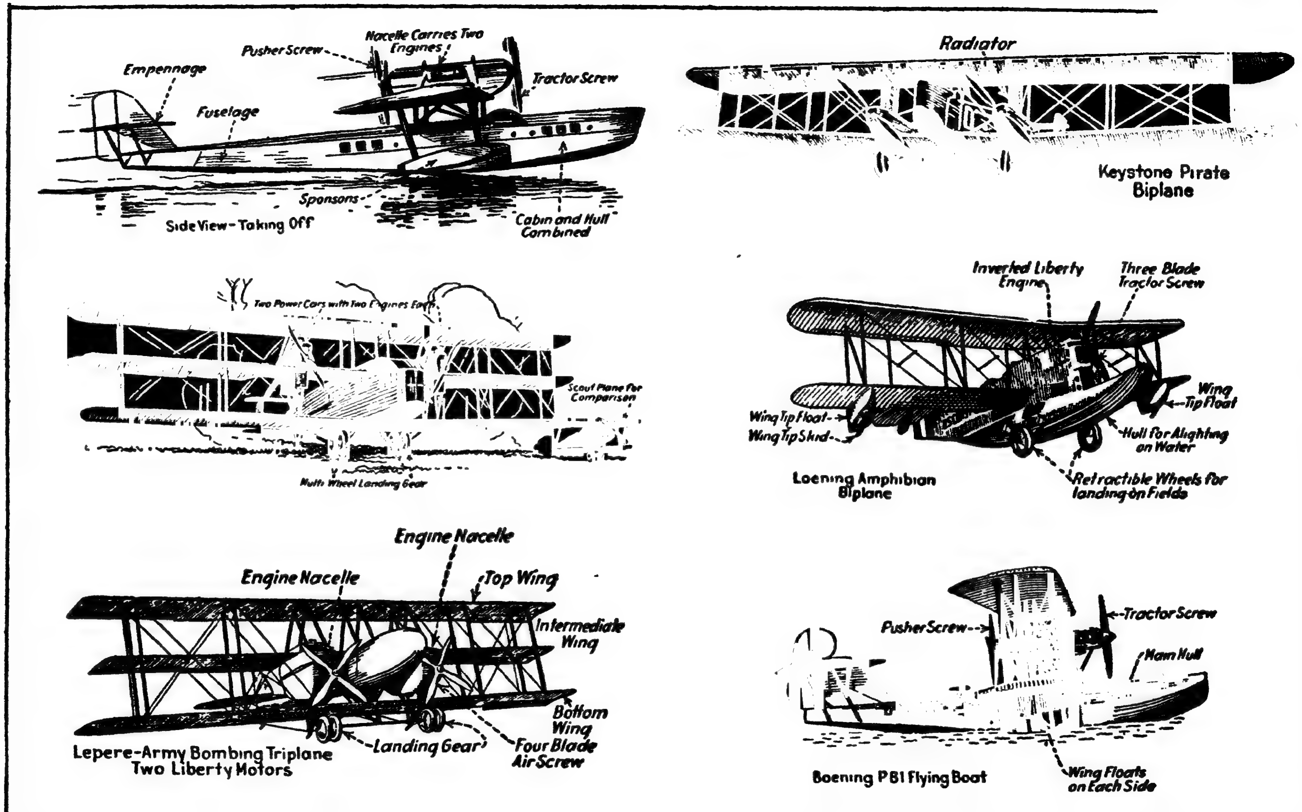


Fig. 1.—Views of Typical Airplanes Showing How Greatly They Vary in Design.

MODERN AVIATION ENGINES

CHAPTER I

AVIATION ENGINE REQUIREMENTS

Brief Consideration of Aircraft Types—Monoplane vs. Biplane—Number of Engines to be Used—Bi-Motor Planes—Tri-Motor Planes—Multi-Motor Idea Not New—Propellers Are Limiting Factor to Engine Size—Essential Requirements of Aerial Motors—Aviation Engines Must be Light—Factors Influencing Power Needed—Resistance to Flight—Formula to Find Horsepower Needed for Flight—Power Used by Airplanes—Why Explosive Motors Are Best—Wet or Dry Weights—History of Engine Development—Main Types of Internal Combustion Engines—Classification by Cylinder Arrangement—Weight-Horsepower Ratio—Engine Types Defined—Life of Aviation Engines—Future Engine Development—Airplane Engine Costs.

The conquest of the air is one of the most stupendous achievements of the ages. Human flight opens the sky to man as a new road, and because it is a road free of all obstructions and leads everywhere, affording the shortest distance to any place, it offers to man the prospect of unlimited freedom. The aircraft promises to span continents like railroads, to bridge seas like ships, to go over mountains and forests like birds, and to quicken and simplify the problems of transportation. While the actual conquest of the air is an accomplishment now being realized, the yearning to conquer the air is old, possibly as old as intellect itself. The myths and folklore of different races tell of winged gods and flying men, and show that for ages to fly was the highest conception of the sublime. No other agent is more responsible for sustained flight than the internal combustion motor, and it was only when this form of prime mover had been fully developed that it was possible for man to leave the ground and alight at will, not depending upon the caprices of the winds or lifting power of gases as with the balloon. It is safe to say that the solution of the problem of flight would have been attained many years ago if the proper source of power had been available as all the essential elements of the modern airplane and dirigible balloon, other than the powerplant, were known to early philosophers and scientists, though in much cruder forms.

Brief Consideration of Aircraft Types.—Aeronautics is divided into two fundamentally different branches—aerodynamics and aerostatics. The first comprises all types of airplanes and heavier-than-air flying machines such as the helicopters, ornithopters, etc.; the second includes dirigible balloons, passive balloons and all craft which rise in the air by utilizing the lifting force of gases. Airplanes are the only practical form of heavier-than-air machines, as the helicopters (machines intended to be lifted directly into the air by propellers, without the sustaining effect of planes), and ornithopters, or flapping wing types, have not been thoroughly developed, and in fact, there are so many serious mechanical problems to be solved before either of these types of aircraft will function properly that experts express doubts regarding the early perfection of either. Airplanes

are divided into two main types—monoplanes or single surface forms, and biplanes or machines having two sets of lifting surfaces, one suspended over the other. A third type, the triplane, is not very widely used at the present time, though several forms are illustrated that have been used in

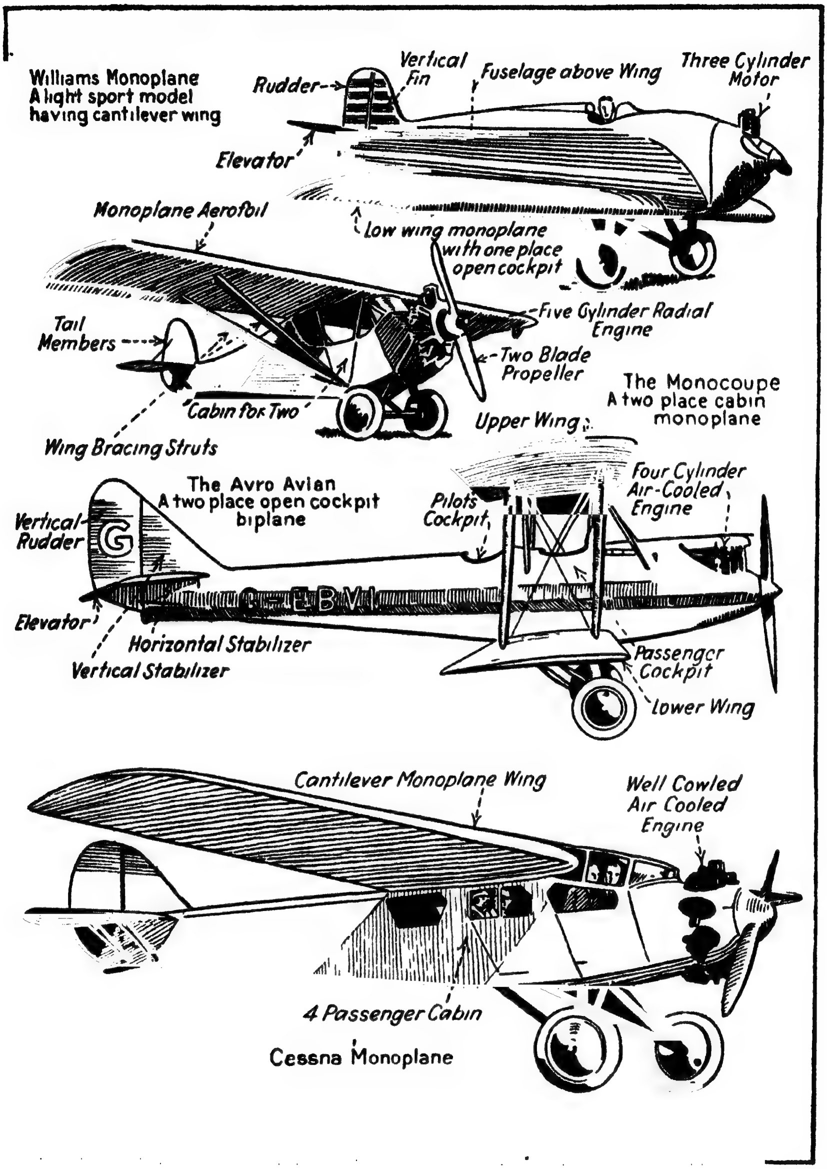


Fig. 2.—Various Forms of Sporting Type Airplanes Using Low-Powered Engines.

a practical way. Airplanes have been built in many types and sizes and for many purposes. The great variety of practical designs available are

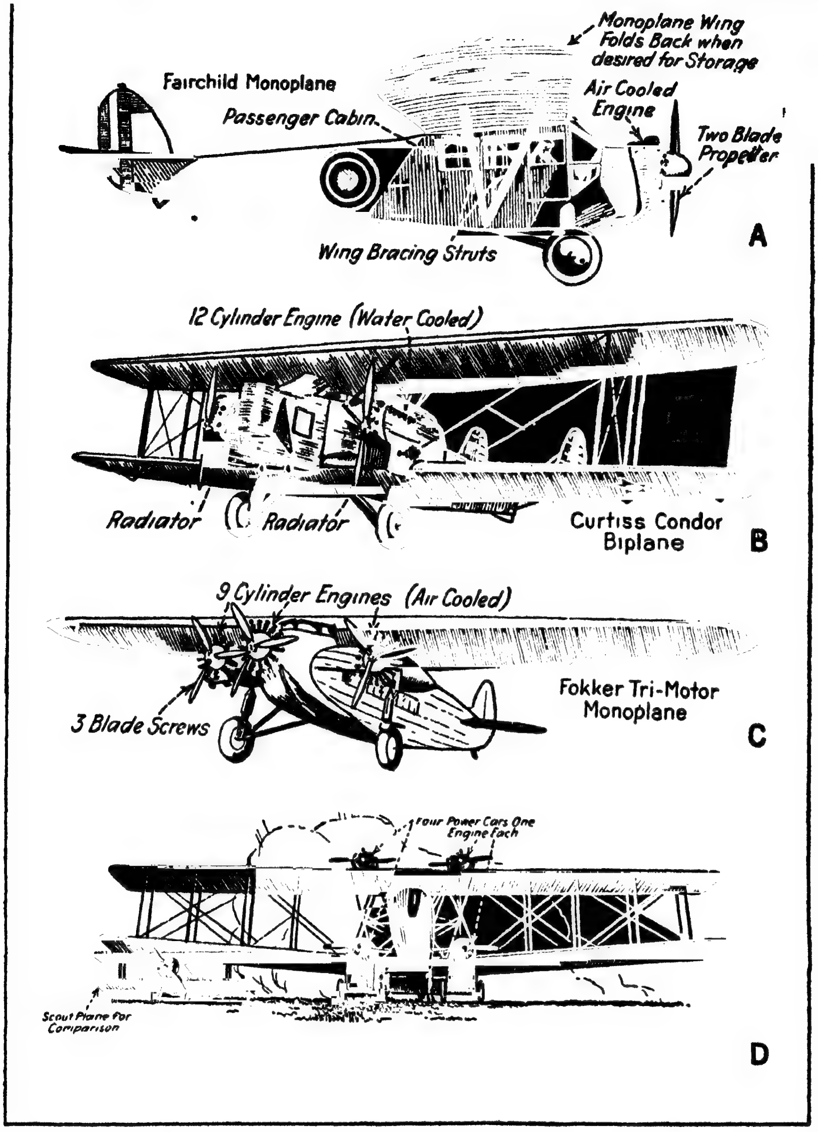


Fig. 3.—Illustration Showing How the Number of Engines Used May Vary. A—Single Motor Monoplane. B—Bimotor Airplane of the Biplane Design. C—Tri-motor Passenger Carrying Monoplane. D.—Bleriot Airliner Using Four Motors.

shown at Fig. 1, some of which are adapted for alighting only on land, others only on water and one type, the amphibian, that can land and take-off from either land or water. The cuts at Fig. 2 show single engine airplanes in a variety of designs and illustrations at Figs. 3 and 4 show multi-motored designs that have received practical applications.

Dirigible balloons are divided into three classes: the **rigid**, the **semi-rigid**, and the *nonrigid*. The *rigid has a frame or skeleton of either wood* or metal inside of the bag, to stiffen it; the semirigid is reinforced by a longitudinal keel and metal attachments; while the nonrigid is just a bag filled with gas. The various types are shown at Fig. 5. The airplane, more than the dirigible and balloon, stands as the emblem of the conquest of the air. Two reasons for this are that power flight is a real conquest of

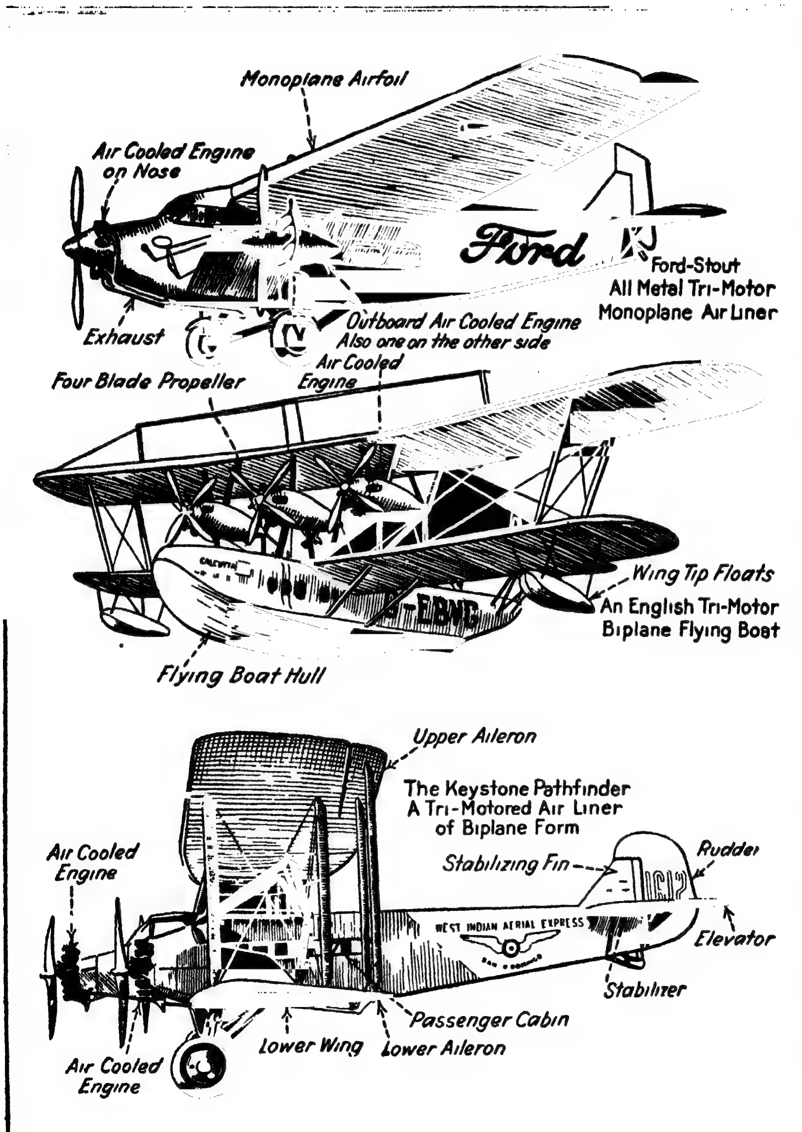


Fig. 4.—Practical Forms of Trimotor Monoplane and Biplane Airliner.

the air, a real victory over the battling elements; secondly, because the airplane, or any flying machine that may follow, brings air travel within the reach of everybody. In practical development, the dirigible may be the steamship of the air, which will render invaluable services of a certain kind, and the airplane will be the automobile of the air, to be used by the multitude, perhaps for as many purposes as the automobile is now being used.

• Considering the helicopter, a number of experimenters in the United States and Europe are studying the problem and it will ultimately be solved. Helicopters are made in many types, a typical experimental form being shown at Fig. 6. Two lifting screws are used, driven by shafts and gearing, and planes, similar to airplane wings, (not shown in illustration) are also used to supplement the lifting effect of the air screws, when the machine is flying on a horizontal plane. The Cierva Auto-giro is not a helicopter but an airplane having a revolving four vaned supporting surface, not driven by the engine directly but turned by the movement of the structure through the air.

Monoplane vs. Biplane.—Study of the designs for transport airplanes in Europe and in the United States indicates that a time-honored debate is still in progress as to the advantage of the monoplane over the biplane, or vice versa. English designers have favored biplanes almost exclusively. In France, one type seems to be as much in favor as the other and some makers, such as H. and M. Farman build both types for the same purpose, while in Germany the monoplane seems to predominate, although some very capable German designers adhere to biplane construction.

The choice between the two types of construction seems to be largely a matter of personal preference, as well as the particular type of work to be accomplished. Aerodynamically, those favoring the biplane claim there is but little to choose between the two, provided the biplane is properly designed though it is conceded that it is easier to design an aerodynamically efficient monoplane than it is to design a biplane of equal efficiency. Chief advantages of the monoplane over the biplane are:

(1) Greater simplicity, as it has fewer parts
(2) The highwing monoplane affords better visibility for passengers
(3) The wing of a highwing monoplane is less liable to damage from obstacles on the ground, such as fence posts and stumps.

The advantages of the biplane are:

(1) The wing cellule, in general, has less weight per square foot than the monoplane
(2) The wing structure is more rigid, and errors in rigging are more easily corrected
(3) The wings can be made up in sections and are more easily handled and stored.

As to the greater simplicity claimed for the monoplane, this is more apparent than real. Of course, there is only one wing, but to carry a given load with a given power the same surface is necessary in either type and the monoplane wing must be of larger area than either of the wings of a biplane are. This means that its internal structure is much more complex than it appears to be at first glance.

The subject of biplane vs. monoplane has been given careful consideration in various books on airplane designs or construction and the interested reader is referred to *Modern Aircraft*, a companion volume to this treatise for a complete elementary exposition of aerodynamical principles, types of airplanes and their advantages and other pertinent matter that is out of place in a discussion on aircraft powerplants.

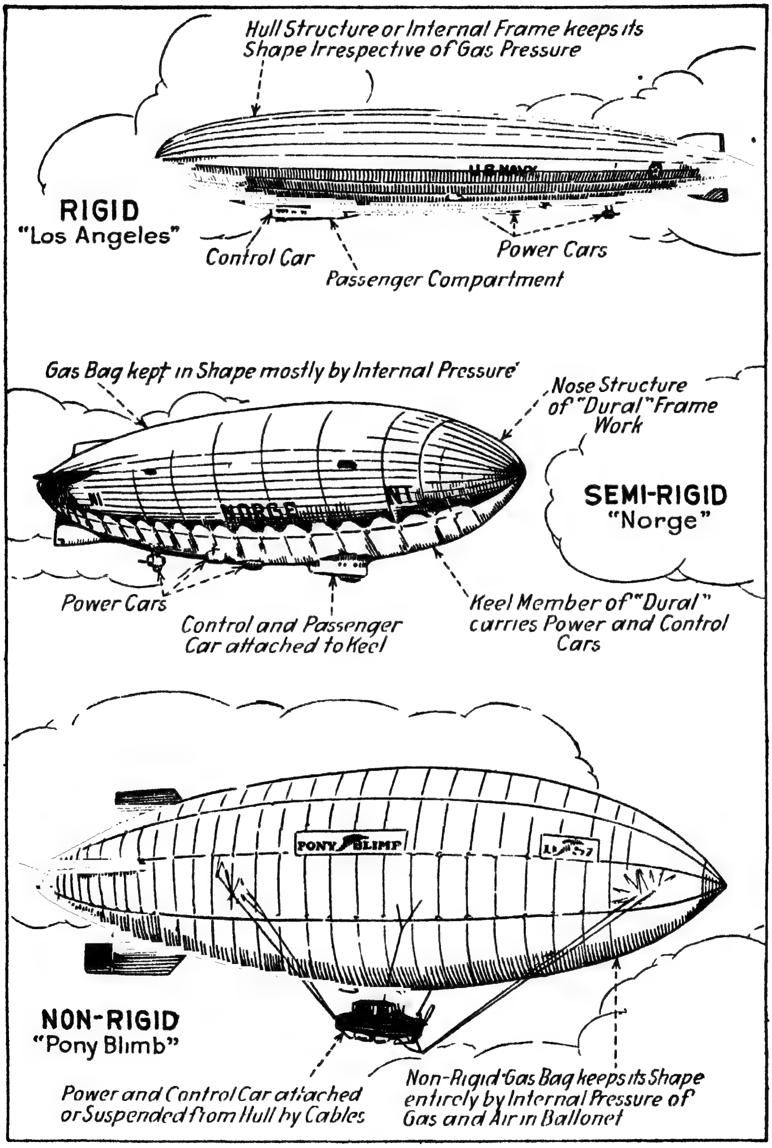


Fig. 5.—Illustration Showing Leading Types of Lighter-than-Air Craft. That Shown at the Highest Altitude is the Rigid Type; the One Flying Below it is a Semirigid, While That Near the Ground is a Nonrigid Dirigible Commonly Called a "Blimp."

Number of Engines to be Used.—A matter that is being given considerable attention and that should be briefly touched upon before considering engine types and their care and repair is the number of engines to be used for power. An authority on multi-engine airplanes, Mr. A. H. G. Fokker has covered this subject very well in the *S. A. E. Journal* and has corrected several erroneous impressions that have prevailed in the past relative to the safety features and efficiency of multi-engine airplanes as compared to the

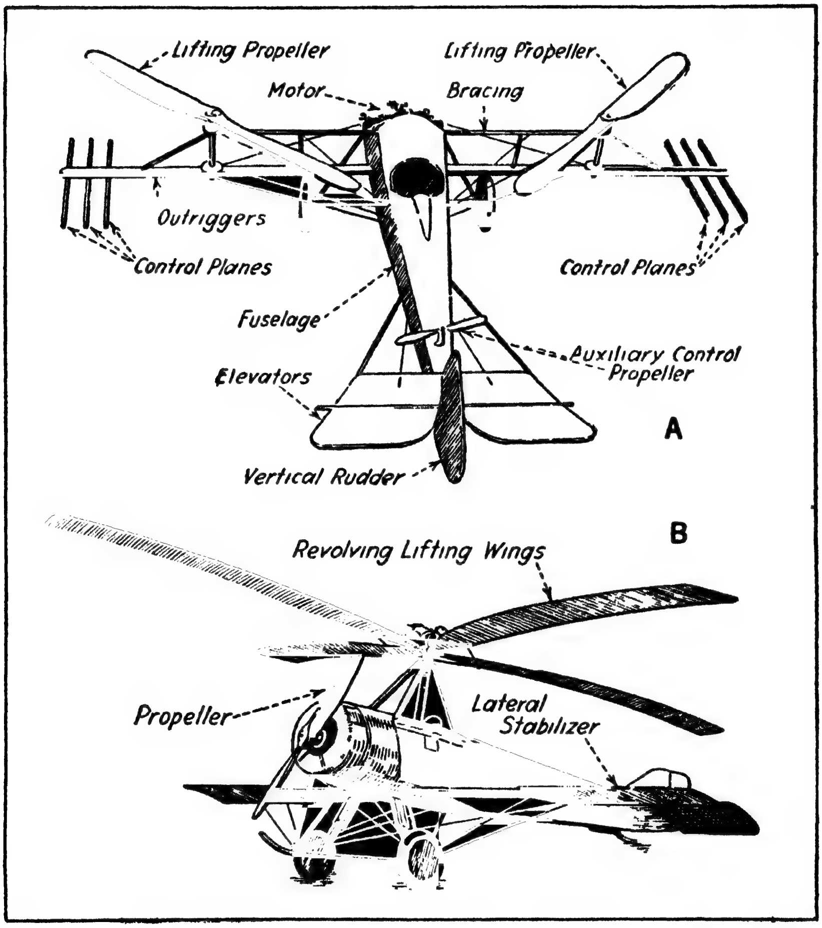


Fig. 6.—Unusual Forms of Heavier-than-Air Craft. A—Experimental Berliner Helicopter. B—The Cierva Rotating Wing Auto-Giro.

single-engine type. Not only does he consider the matter from a practical viewpoint but he outlines various points in which theory and practice do not agree as well as they should. Mr. Fokker says:

"The facts that different kinds of multi-engine airplanes have been developed for a long time, although mostly for military purposes, and that during the last few years such airplanes have been in use for commercial operation, have frequently led the public to form the opinion that a multi-

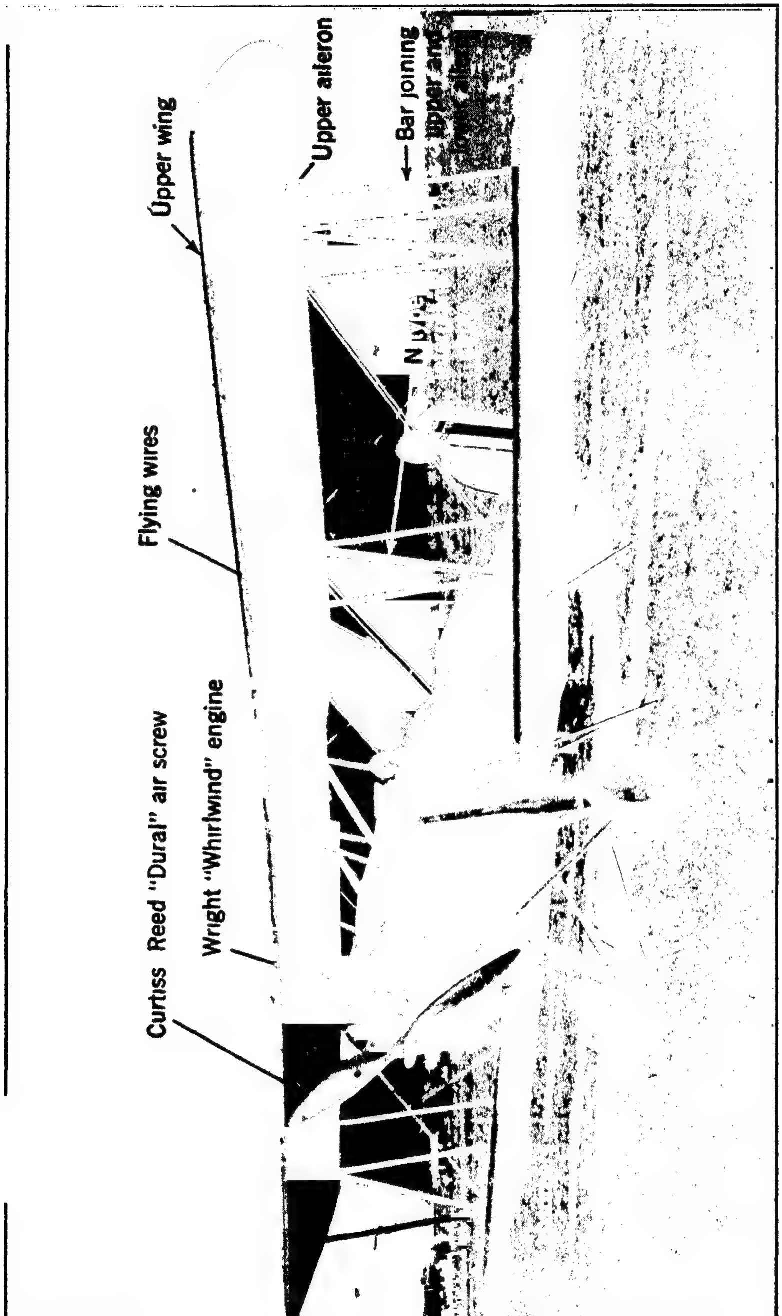


Fig. 7.—Curtiss Airplane for Student Instruction with Wright "Whirlwind" Motor Installed at Front End of the Fuselage.

engine craft is safer than a single-engine craft. On many European airlines many multi-engine airplanes have been and are still in operation in which the increased horsepower is utilized for no other reason than to increase their carrying capacity, rather, than to obtain safety through their ability to fly on the remaining engine or engines in the event of the failure of one powerplant. Multi-engine airplanes, therefore, can be divided into two classes:

(1) Those able to continue flight with the full normal load after one or more of the engines have stopped

(2) Those not able to continue flight with the full normal load after one or more of the engines have stopped.

"Class (2) can be subdivided according to the reasons for which more than one engine is used; namely,

(1) Those used for military purposes, as for bombing or similar special purposes, to obtain a fuselage having a nose without an engine

(2) Those in which the engine power required by the design and by the purpose for which the airplane is built is not available in one powerplant, two or more small engines being used instead of one large one.

"It is obvious that in Class (2) the actual risk of a forced landing is multiplied by the number of powerplants used, and therefore the reliability decreases with the number of engines.

"Until a short time ago, multi-engine airplanes in Class (1) did not exist. Even now I am risking protest from several designers by stating that two-engine planes in Class (1) practically do not exist. I know of many newly designed planes that have demonstrated their ability to fly with one engine on their first test flight; but an enormous difference exists between the demonstration of a new well-tuned plane and the same type later on when in production, really fully equipped, and after all the reinforcements, improvements and additions, which never fail to be made after the first plane has been demonstrated, have been incorporated into it. Certainly, after such planes have been in service for some time, the efficiency drops because of the wear and tear on the engines and because the finish of the ship and the propellers have likewise suffered from the weather.

"**Bimotor Planes.**—The ability of a two-engine airplane to fly on one engine is proved only if such a craft is able not only to fly for a short time on one engine, but is able to develop sufficient reserve to run on one engine for at least a few hours without overheating or overstraining the engine. I have built and have demonstrated several types of two-engine airplanes, but to enable them to fly for a long period under the conditions mentioned would necessitate a reduction of the pay-load to such an extent that the airplane could not be operated economically. Two-engine airplanes used for military purposes, when the excess load can be dropped at any time if it is necessary to continue on one engine, are an entirely different and reasonable proposal, because the carrying capacity can be utilized to limit; whereas the limit of the capacity of a smaller commercial plane, from which it is not feasible to drop the passengers or their luggage, or both, is dictated by the carrying capacity that can be developed with one engine while leaving sufficient reserve for safe flying and maneuvering. It is still very

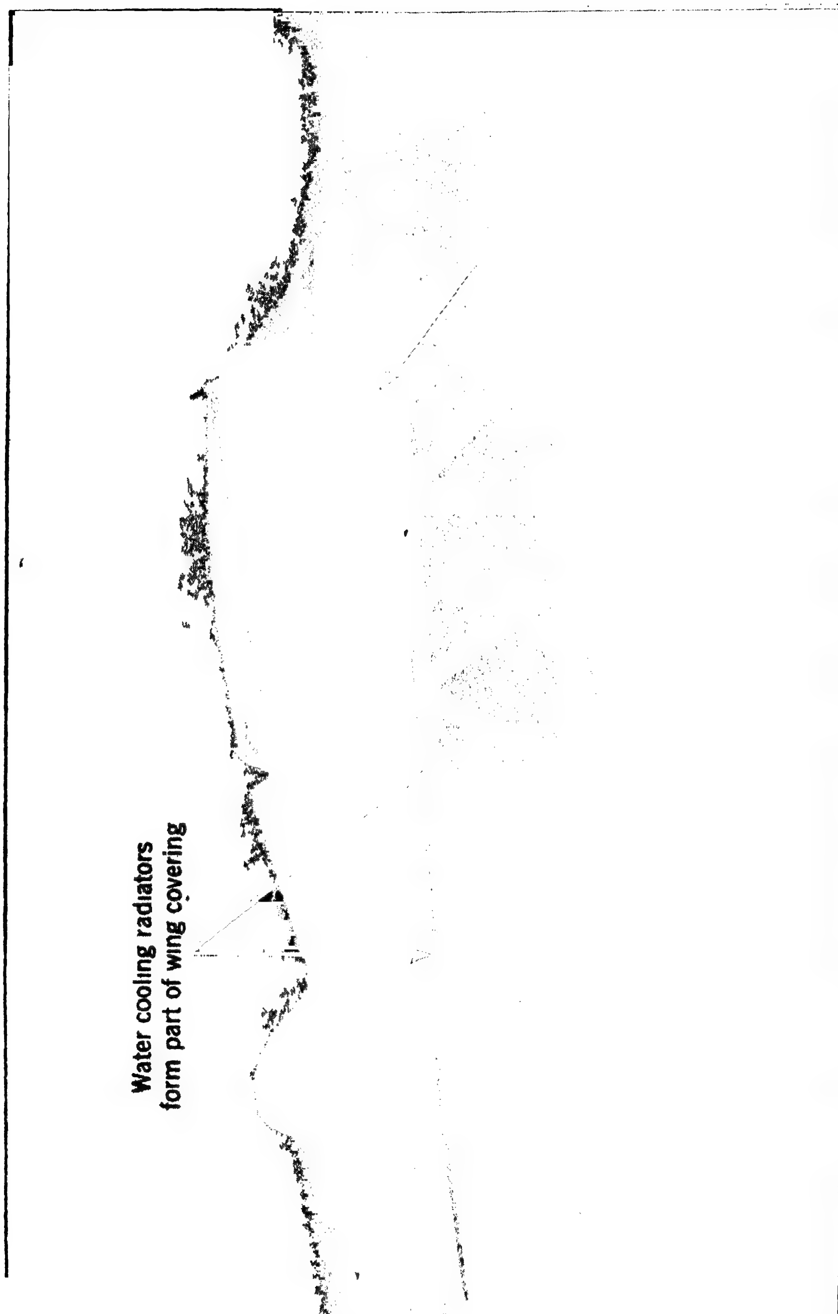


Fig. 8.—Illustration of Macchi Racing Monoplane Showing How Large Water-Cooled Powerplant is Installed in Fuselage, and Covered in by Sheet Metal Streamlining to Reduce Air Resistance.

difficult to produce even three-engine planes that are really and honestly in Class (1) while carrying their normal full commercial load.

"The following is a review of a few of the inherent features that must be borne in mind and that present the difficulties to be encountered in designing and using aircraft that are able to continue safely with one engine dead. Many persons who have never analyzed the matter have the delusion that if one engine of a two-engine plane stops in flight, 50 per cent of the original horsepower is left for the purpose of flight; and, similarly, on a three-engine plane, two-thirds of the original power remains. Unfortunately, this is far from the truth; the facts greatly increase the disadvantages of the two-engine plane in this respect as compared with a three-engine plane. When one powerplant stops, the airplane slows down and the remaining propeller is not able to develop the same number of revolutions as when both the powerplants were driving the airplane at full speed.

"Trimotor Plane.—This applies also to the three-engine airplane. To take a concrete example, without going into mathematics: If the Fokker trimotor plane has propellers that will turn at 1800 r.p.m. at 220 hp. at full speed with all three engines going, the remaining two propellers, if one of the three stops, will turn at only a little more than 1700 r.p.m. Therefore, we have the effect of a total of only 370 instead of 440 hp., as the failure of one of the three engines would lead us to believe. With a two-engine plane the situation is much worse. In a plane having 600 hp. available from two engines, not more than 40 per cent, or 240 hp., would be available from one engine after the other had stopped.

"To this disadvantage of the two-engine plane a further and equally serious one must be added. I refer to planes of the usual type having outboard engines, as very few remarks apply to possible, but hitherto not developed, types having central engine-rooms from which multiple powerplants drive a single propeller. The turning forces applied to the airplane, after the failure of one outboard powerplant are naturally much greater in the two-engine airplane than in one having three engines.

"Disadvantages of Two Engines.—To overcome these disadvantages, correspondingly greater forces must be applied by the controlling surfaces, of whatever design these may be or however they may be disposed, to keep the airplane on its straight path or to make a turn to the opposite direction. Such controlling forces are, so to speak, only the products of head resistance. The two-engine plane therefore suffers a greater increase in head resistance, due to these controlling forces, than does the three-engine plane and requires more power to keep it in the air. This means only that it consumes a greater percentage of the remaining but already reduced horsepower. This is proved by the fact that most designers of two-engine planes are expending considerable ingenuity on special control-surfaces, such as duplicate rudders, emergency rudders, and the like, arranged in positions where they will have the benefit of the slipstream of the remaining propeller and consequently will have a better controlling effect with less expenditure of head resistance. Such devices, however, have other disadvantages, such as increased head resistance at all speeds in normal use and, in the case of military planes, reduction of the field of fire to the rear and greatly increased vulnerability from the enemy's fire.

"But the greatest factor in increasing the inefficiency of the airplane when flying with one engine idle is that the remaining horsepower is further absorbed in overcoming the resistance of the stopped engine and its propeller, *which means that only a small part of the original available horsepower remains to keep the ship in the air. What this percentage is depends* entirely upon the type of the airplane and the location of the defective engine.

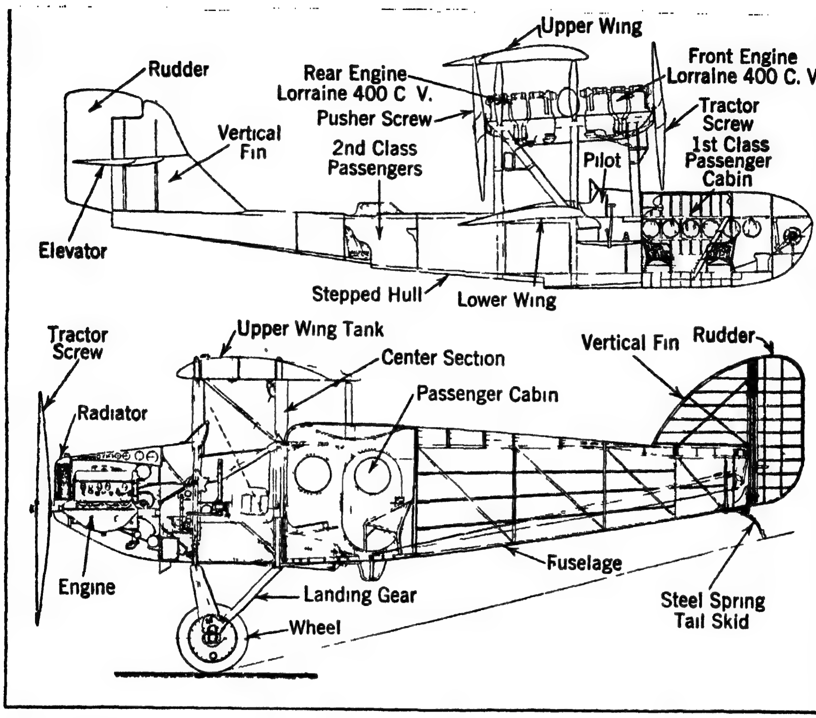


Fig. 9.—Sectional Views of Typical Airplanes, Showing Method of Powerplant Installation. Above, the Nieuport-Macchi Passenger Carrying Flying Boat Equipped with Two Lorraine Twelve-Cylinder Motors. Below, the Nieuport-Delage Biplane with Single Hispano-Suiza Engine Installed at Front End of Fuselage.

"In the arrangement of two powerplants in use on some flying boats, in which one powerplant is located behind the other, the disadvantage due to the turning forces when one engine stops is nonexistent. However, the loss of efficiency of the propellers, with the resulting reduction in speed, and the increased head-resistance of the nondriven propeller and its engine, provide the same problem as that of any other two-engine airplane.

"The multi-engine principle must be considered as insurance against a forced landing due to the failure of one powerplant. This insurance, of course, cannot be obtained for nothing; it costs more in fuel consumption and in head resistance, if the insurance is to be good. Therefore, intrinsically, a single-engine ship having the same design and power, so far as possible, is bound to have a longer range. To use the Fokker F-VII again

as an illustration, this trimotor plane, having three Wright Whirlwind engines and the large wing, has a range of about 4,300 miles as compared with the same ship fitted with a single Pratt & Whitney Hornet or a Bristol Jupiter engine, which gives it a range of about 5,500 miles. In simplicity, efficiency, economy of running, and initial cost, the advantages are all on the side of the single-engine plane."

At the present stage of development, the weight per horsepower of an engine does not actually vary in proportion with the power. That is to say, present day low powered engines weigh per horsepower, more than the larger engines. The effect of this upon modern design will be that a plane requiring a given power, if it is to incorporate the three-engine principle, will have a smaller amount of pay-load carrying capacity than a similar single-engine plane of equal horsepower, though this deficiency will probably not be very great. Aerodynamical considerations favor a single engine because it is obvious that it will offer less parasitic resistance than three independent engines. On the other hand, the almost complete freedom from forced landings will be very important to air transport and, in fact, to all forms of aerial service. It seems safe to predict, therefore, that the three-engine airplane will be an absolute necessity to the successful operation of a passenger carrying air transport project. In such a case a passenger airliner will be equipped with three small engines preferably air-cooled, together totaling the required maximum horsepower necessary for taking-off and climbing, or in making headway against the highest head-winds which might be met with.

Under normal conditions the plane would be flown on two-thirds or three-quarters throttle for all engines, just as in a single-engine plane, but in the event of failure of any one engine, the power of the other two at full throttle will be sufficient to maintain level flight until the plane reaches its destination. The parasitic resistance that obtains when three radial cylinder air-cooled engines are used for power can be greatly reduced by properly streamlining the engine supports, especially those of the outboard engines, as properly housing the engine mounted in the fuselage offers no particular difficulty. The outboard engines of the Fokker airplane are carried below the monoplane wing structure. The engine is almost entirely enclosed in a metal cowling and only the cylinder heads and upper portion of the cylinders project into the air stream.

A giant new Dornier plane which, it is said, will be powered by twelve Bristol Jupiter engines of 500 hp. each, is reported under construction on a special wharf at Lake Constance. The craft is designated as the Dornier 10. Early reports of the specifications put the fuselage length at 40 yd. and the span at 50 yd. The cruising speed of the Dornier 10 will approximate 115 m.p.h. and the maximum speed 149 m.p.h. It is estimated that the cruising range with full load of passengers and freight will be between 1,625 miles and 2,500 miles. The fuselage, of duralumin construction, is being formed in two stories, the lower one to accommodate more than twenty passengers with lounge, sleeping compartments, kitchen, etc. Officers and crew, totalling nine persons, are to navigate the plane from the upper story of the fuselage. Freight is to be stored in the latter section, it is said.

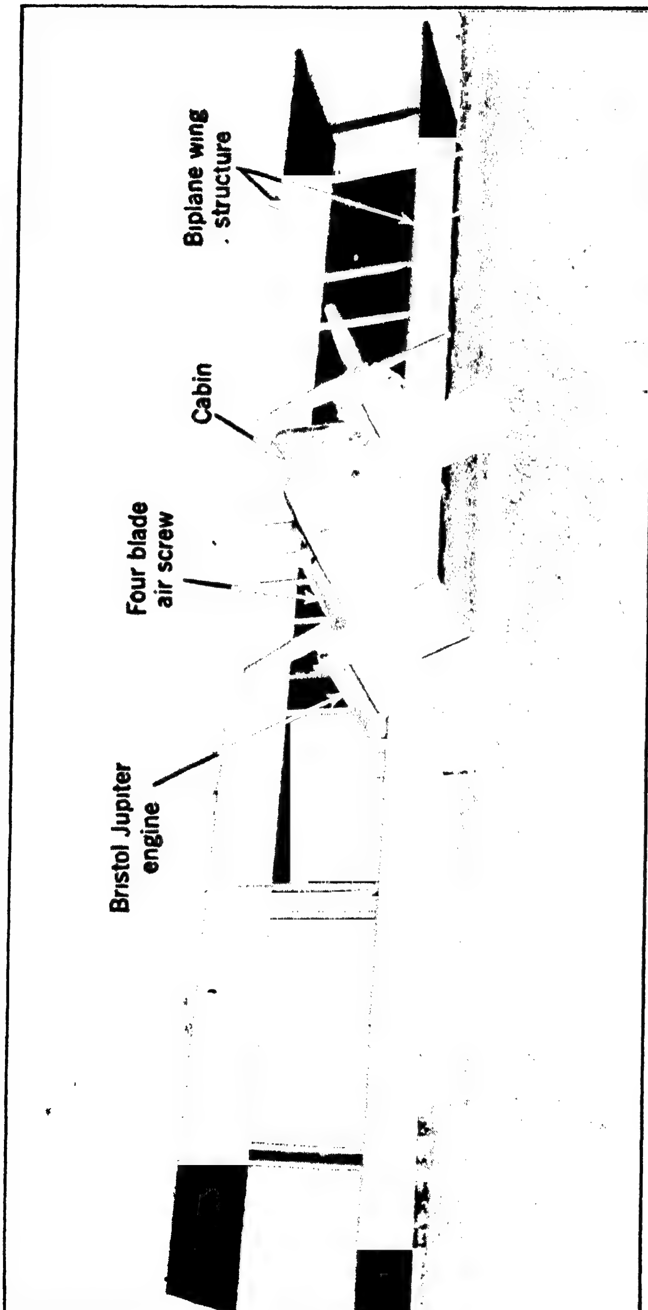


Fig. 10.—Farman-Goliath Biplane with Two Bristol-Jupiter Engines for Power.

Multimotor Idea Not New.—At the outbreak of the war, the idea of constructing planes to be driven by two engines had been broached and discussed, detailed designs had been made for such machines and their powerplant equipment, experiments had been conducted, and several multiple-engined airplanes had been constructed. Nevertheless, it was not until June, 1915, that information was received of the definite appearance at the front of a twin-engined German airplane. Although the Germans were thus, apparently, the first to employ such a machine for active military service, they were probably anticipated as regards actual construction by the French twin-engined Caudron biplane.

The three-engined airplane had also received some attention when the war broke out, notably so from the Italian Caproni, who, in 1914, built and flew a biplane equipped with two 80-hp. tractor engines and a 90-hp. pusher engine. Subsequently, in 1915, the same designer built a successful biplane fitted with three 150-hp. engines and later built large triplanes with three engines. As the war progressed and the demand arose for heavy bombing machines, the twin-engined airplane took a permanent place in the aeronautical services of all the belligerents. Of these, the Gotha with two 260-hp. engines and the Handley-Page with two 350-hp. units may be taken as typical.

Toward the end of the war the four-engined machine had definitely appeared, and was being built in considerable numbers, in England, at least, while a five-engined German machine was brought down in France in August, 1918. Since the armistice was signed, the four-engined machine has become quite familiar, and the development of the five-engined design has progressed to the extent that there is now in existence a large British seaplane equipped with five Rolls-Royce engines which has flown successfully. Reports indicate that airplanes with eight engines are to be built in England and ten- and twelve-engine seaplanes of very large dimensions are projected in Germany.

The airplane powerplant problem at the present moment is a peculiarly complicated one, and in no respect is its complexity greater than in multiple-engined machines. A very strong reason for fitting an airplane with more than one engine, at least so far as civilian flying is concerned, is, of course, the increased safety insured by so doing. With a twin-engined machine the chances of both failing before a safe landing can be effected are now very small. During the war, it may be said by way of illustrating this remark, a Handley-Page bomber as a result of a direct hit had its lower wings reduced to shreds and tatters and one of its engines put out of action; yet this machine flew back 60 miles to its airdrome, and lighted there safely. With a three-engined machine, it may be taken that all chances of having to make a forced landing as a result of engine trouble developing are eliminated. A four-engined airplane possesses the same characteristic, and with something over. Indeed, it can be asserted that as regards the avoidance of forced descents the four-engined machine possesses a factor of safety which is, or should be, satisfactory to all concerned. To increase the number of engines above four on the grounds of safety only is clearly superogatory, the real reason is the need for more power.

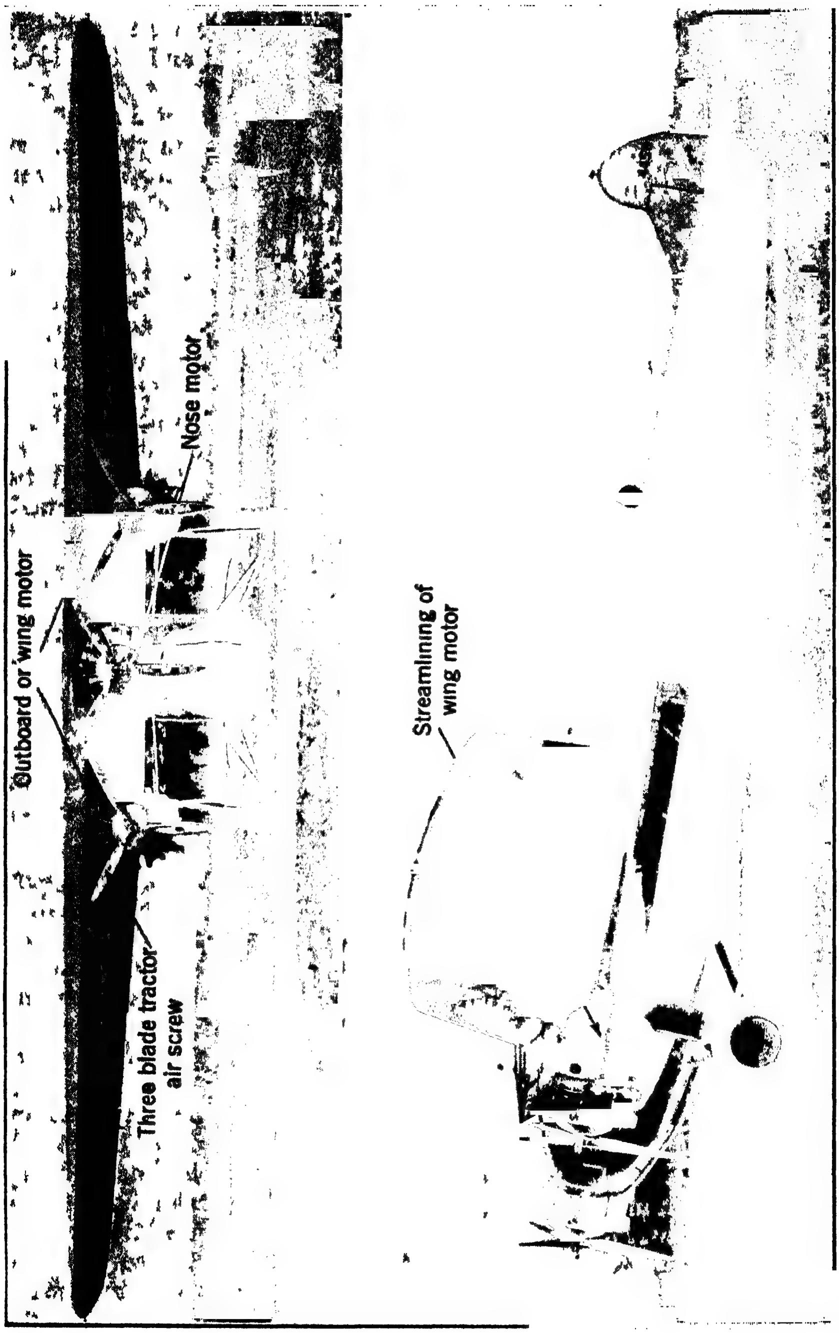


Fig. 11.—Views of Fokker Trimotor Passenger Carrying Monoplane with Three Radial Air-Cooled Engines Installed. Note Method of Locating Wing Motors and Use of Three Blade Air Propellers.

Setting aside such cases, if there be any, in which the multiplicity of engines is dictated by a desire to utilize existing stocks left over from the war program, it might be suggested that several small units are preferred to one or two larger units because the design and production of the former have been brought to a considerable degree of perfection, whereas the large aircraft engine possessing an equal trustworthiness has yet to be built. So far as air-cooled engines are concerned, it may be true that large units are not installed because large units are not yet available. On the other hand, the largest size at present made commercially, we have recently seen an air-cooled engine developing 500 b. hp., is sufficient to effect a reduction of some 25 per cent in the number of engines fitted on the four-engined

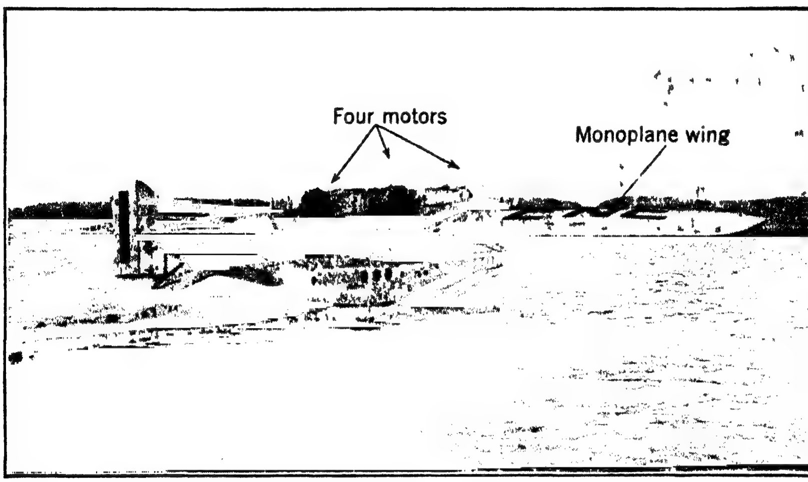


Fig. 12A.—Dornier Super-Wal Monoplane Flying Boat with Four Air-Cooled Engines Mounted Above Aerofoil.

Handley-Page machine. Water-cooled engines of 500 hp., such as the eighteen-cylinder, three-row Sunbeam, have been manufactured in England for some time, while single engines of 1,000 hp. and even 1,250 hp. are within reach of present-day production. In support of the latter assertion, it may be said that a twenty-four-cylinder Liberty engine has, under special conditions, developed over 800 hp., and that a twenty-four-cylinder engine of Packard design has been made recently and tested with satisfactory results that was of over 1,000 hp. though it did not produce twice the power of a twelve-cylinder of one-half the displacement.

Propellers Limiting Factor to Engine Size.—It seems clear, then, that the tendency to multiply the engine units on an airplane cannot be set down wholly to a deficiency of large powered engines. The true reason, or a large part of it, for adopting a multiplicity of engines lies, in fact, not with the aircraft engine builder, but with the makers of the airplane itself and of its propellers. The eight-engined seaplane referred to will probably have a horsepower of nearly 3,000. Even were a thoroughly trustworthy 1,000-hp. aeronautic engine available, it is doubtful if in the present state of the aircraft building art the designers would have chosen to do other than em-

ploy eight small engines rather than three large; for by splitting up the power between a number of units, they effect a corresponding distribution of the flying stresses in the structural parts of the machine. Further, it can be asserted that the airplane propeller capable of using 1,000 hp. on a scale of efficiency comparable with that manifested by smaller existing propellers has yet to be designed and made. It is to be noted that, as with the marine propeller, the higher the engine speed the more difficult is it to provide a propeller which will utilize the available power efficiently, and that aeronautic engine builders are already sacrificing something to meet the short-comings of the propeller by fitting reduction gearing on their higher speed engines. Thus the Rolls-Royce, Sunbeam, Hispano and Cosmos engines, all of which run at or over 2,000 r.p.m., are forced to employ reduction gearing, of approximately a five to three ratio, because of the present impossibility of obtaining propellers capable of utilizing the full speed economically.

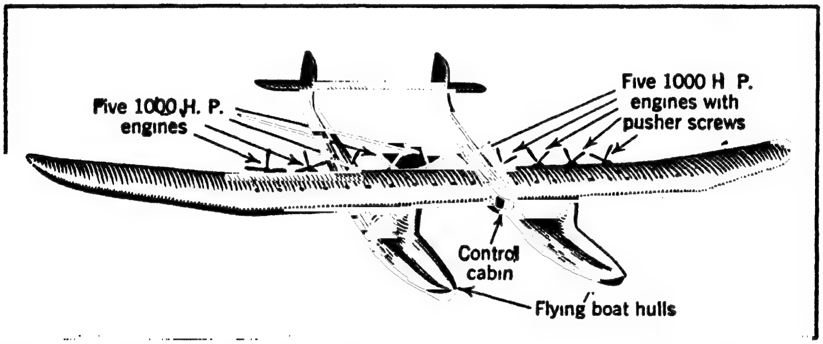


Fig. 12B.—Model of a Giant Air Liner Designed by Dr. Rumpler of Berlin. Ten 1,000 Horsepower Engines are Placed in Trailing Edge of All-Metal Wing, to Contain Passengers in Leading Edge. Note Double Hull Construction.

Essential Requirements of Aerial Motors.—One of the marked features of aircraft development has been the effect it has had upon the refinement and perfection of the internal combustion motor. Without question gasoline-motors intended for aircraft are the nearest to perfection of any other type yet evolved. Because of the peculiar demands imposed upon the aeronautical motor it must possess all the features of reliability, economy and efficiency now present with automobile or marine engines and then must have distinctive points of its own. Owing to the nature of the medium through which it is operated and the fact that heavier-than-air machines can maintain flight only as long as the power plant is functioning properly, an airship motor should be more reliable than any used on either land or water though it is difficult to conceive of greater reliability than is now built into automobile engines. While a few pounds of metal more or less makes practically no difference in a marine motor and has very little effect upon the speed or hill-climbing ability of an automobile, an airplane motor must be as light as it is possible to make it because every pound counts, whether the motor is to be fitted into an airplane or in a dirigible balloon.

Aircraft motors, as a rule, must operate constantly at high speeds in order to obtain a maximum power delivery with a minimum piston displacement. In automobiles, or motor boats, motors are not required to run constantly at nearly their maximum speed. Most aircraft motors must function for extended periods at speeds of about three-quarters the maximum and very often must be flown for hours at wide open throttle. Another thing that militates against the aircraft motor is the more or less unsteady foundation to which it is attached. The necessarily light framework of the airplane makes it hard for a motor of ordinary design to perform at maximum efficiency on account of the vibration of its foundation while the craft is in flight. Marine and motor car engines, while not placed on foundations as firm as those provided for stationary powerplants, are installed on bases of much more stability than the light structure of an airplane. The aircraft motor, therefore, must be balanced to a nicety and must run steadily under the most unfavorable conditions.

Aviation Engines Must be Light.—The capacity of light motors designed for aerial work per unit of mass is surprising to those not fully conversant with the possibilities that a thorough knowledge of proportions of parts and the use of special metals developed by the automobile industry make possible. Activity in the development of light motors was more pronounced in France before the war than in any other country but this is not true at present as very light and powerful engines have been developed in England and the United States. Some of these motors have been complicated types made light by the skillful proportioning of parts, others are of the refined simpler form modified from current automobile practice. There is a tendency to depart from the freakish or unconventional construction and to adhere more closely to standard forms because it is necessary to have the parts of such size that every quality making for reliability, efficiency and endurance are incorporated in the design. Airplane motors range from two cylinders to forms having fourteen to twenty-four cylinders and the arrangement of these members varies from the conventional vertical tandem and opposed placing to the Vee form or the more unusual radial motors having fixed cylinders. The weight has been reduced so it is possible to obtain a complete powerplant of the radial cylinder air-cooled type that will weigh around 2.5 pounds per actual horsepower and in some cases less than this.

If we give brief consideration to the requirements of the aviator it will be evident that one of the most important is securing maximum power with minimum mass, and it is desirable to conserve all of the good qualities existing in standard automotive motors. These are certainty of operation, good mechanical balance and uniform delivery of power—fundamental conditions which must be attained before a powerplant can be considered practical. There are in addition, secondary considerations, none the less desirable, if not absolutely essential. These are minimum consumption of fuel and lubricating oil, which is really a factor of import, for upon the economy depends the capacity and flying radius. As the amount of liquid fuel must be limited the most suitable motor will be that which is powerful and at the same time economical. Another important feature is to secure accessibility of components in order to make easy repair or adjustment of parts possible.

It is possible to obtain sufficiently light-weight motors without radical departure from established practice. Water-cooled powerplants have been designed that will weigh but 2.5 to 3 pounds per horsepower and in these forms we have a practical powerplant capable of extended operation. Air-cooled engines of recent development have shown remarkable performances and hold many records for endurance and transoceanic flights.

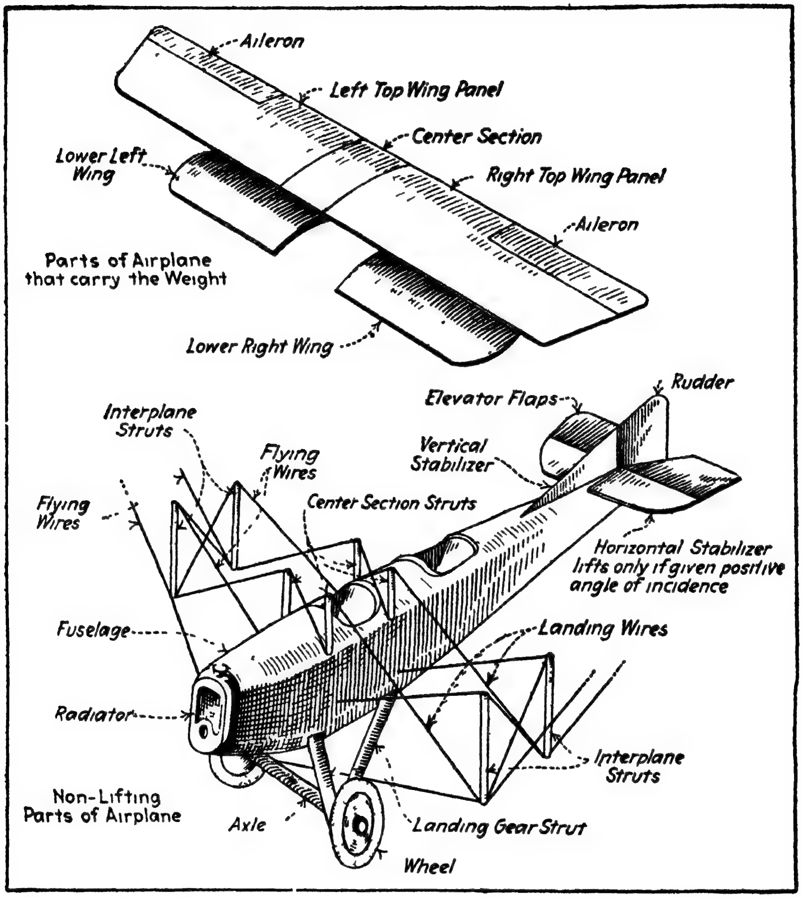


Fig. 13.—Illustrations Showing Parts of an Airplane of the Biplane Form that Offers Resistance, Separated into Main Assemblies. The Wings Shown at the Top Carry the Weight, Because of Air Resistance which Provides Lift Also, but the Parts Shown Below Offer Only Parasitic Resistance which Should be Reduced as Much as Possible by Careful Design to Secure Aerodynamical Efficiency.

Factors Influencing Power Needed.—Work is performed whenever an object is moved against a resistance, and the amount of work performed depends not only on the amount of resistance overcome but also upon the amount of time utilized in accomplishing a given task. Work is measured in horsepower for convenience. It will take one horsepower to move 33,000 pounds one foot in one minute or 550 pounds one foot in one second. The same work would be done if 330 pounds were moved 100 feet in one minute.

It requires a definite amount of power to move any automobile or other vehicle over the ground at a certain speed, so it must take power to overcome resistance which resists forward movement of an airplane in the air. Disregarding the factor of air density, it will take more power as the speed increases if the weight or resistance remains constant, or more power if the speed remains constant and the resistance increases.

The airplane is supported by air reaction under the planes or lifting surfaces and the value of this reaction depends upon the shape of the aerofoil, the amount it is tilted and the speed at which it is drawn through the air. The angle of incidence or degree of wing tilt as well as aerofoil cross section regulates the power required to a certain degree as this affects the speed of horizontal flight as well as the resistance. Resistance may be of two kinds, one that is necessary and the other that it is desirable to reduce to the lowest point possible. There is the wing resistance and the sum of the resistances of the rest of the machine such as fuselage, struts, wires, landing gear, etc. The wing resistance is useful, the other is called parasitic resistance. An illustration showing the parts that carry the weight in an airplane and those that offer parasitic resistance is given at Fig. 13. If we assume that a certain airplane offered a total resistance of 300 pounds and we wished to drive it through the air at a speed of sixty miles per hour, we can find the propeller horsepower needed by a very simple computation as follows:

The product of

$$\frac{\text{300 pounds resistance times speed of 88 feet per second times 60 seconds in a minute}}{\text{divided by 33,000 foot pounds per minute in one horsepower}} = \text{HP. needed.}$$

The result is the horsepower needed in the form of propeller pull

$$\frac{300 \times 88 \times 60}{33{,}000} = 48 \text{ HP.}$$

Just as it takes more power to climb a hill than it does to run a car on the level, it takes more power to climb in the air with an airplane than it does to fly on the level because the resistance increases. It also requires more power to fly fast than it does at lower speeds. The more rapid the climb, the more power it will take.

If the resistance remains 300 pounds and it is necessary to drive the plane at 90 miles per hour, we merely substitute proper values in the above formula and we have

$$\frac{\text{300 pounds times 132 feet per second times 60 seconds in a minute}}{\text{33,000 foot pounds per minute in one horsepower}} = 72 \text{ HP.}$$

The same results can be obtained by dividing the product of the resistance in pounds times speed in feet per second by 550, which is the foot-pounds of work done in one second to equal one horsepower. Naturally, the amount of propeller thrust measured in pounds necessary to drive an airplane must be greater than the resistance by a substantial margin if the

plane is to fly and climb as well. The following formulæ were given in *The Aeroplane* of London and can be used to advantage by those desiring to make computations to ascertain power requirements:

The thrust of the propeller depends on the power of the motor, and on the diameter, pitch of the propeller and its speed of rotation. If the required thrust to a certain machine is known, the calculation for the horsepower of the motor should be an easy matter.

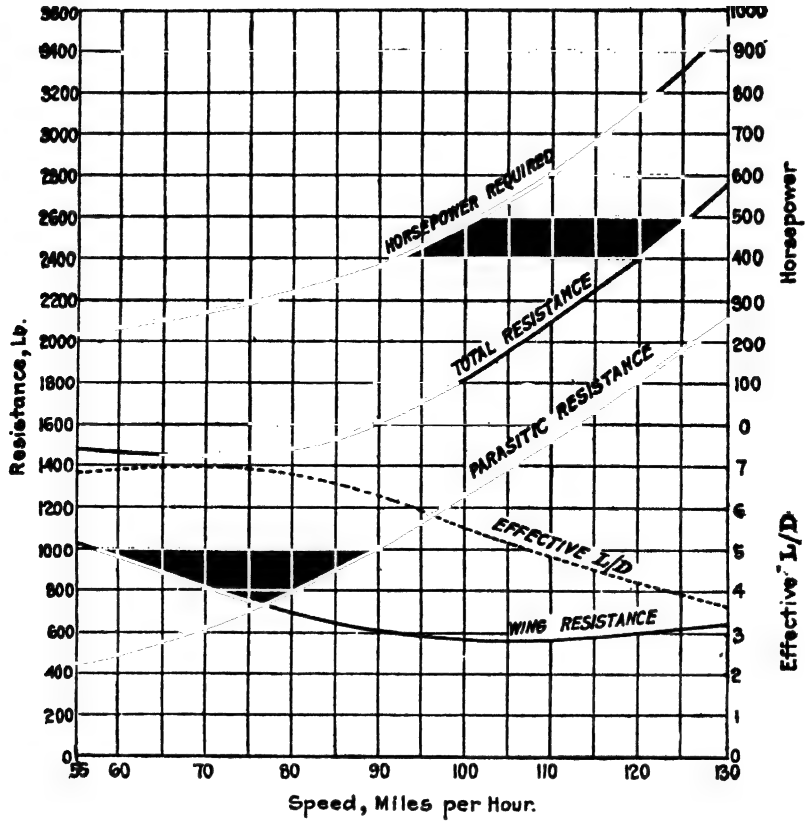


Fig. 13A.—How Various Resistance Values and Speeds Affect Amount of Horsepower Required for Flight.

Resistance to Flight.—The required thrust is the sum of three different "resistances." The first is the "drift" (dynamical head resistance of the aerofoils), i.e., tan $\alpha \times$ lift (L), lift being equal to the total weight of machine (W) for horizontal flight and α equal to the angle of incidence. Certainly we must take the tan α at the maximum K_y value for minimum speed, as then the drift is the greatest (Fig. 14 A).

Another method for finding the drift is $D = K \times AV^2$, when we take the drift again so as to be greatest.

The second "resistance" is the total head resistance of the machine, at its maximum velocity. And the third is the thrust for climbing. The horsepower for climbing can be found out in two different ways. It is first proposed to

deal with the method, where we find out the actual horsepower wanted for a certain climbing speed to our machine, where

$$\text{HP.} = \frac{\text{Climbing speed/sec.} \times W}{550}$$

In this case we know already the horsepower for climbing, and we can proceed with our calculation.

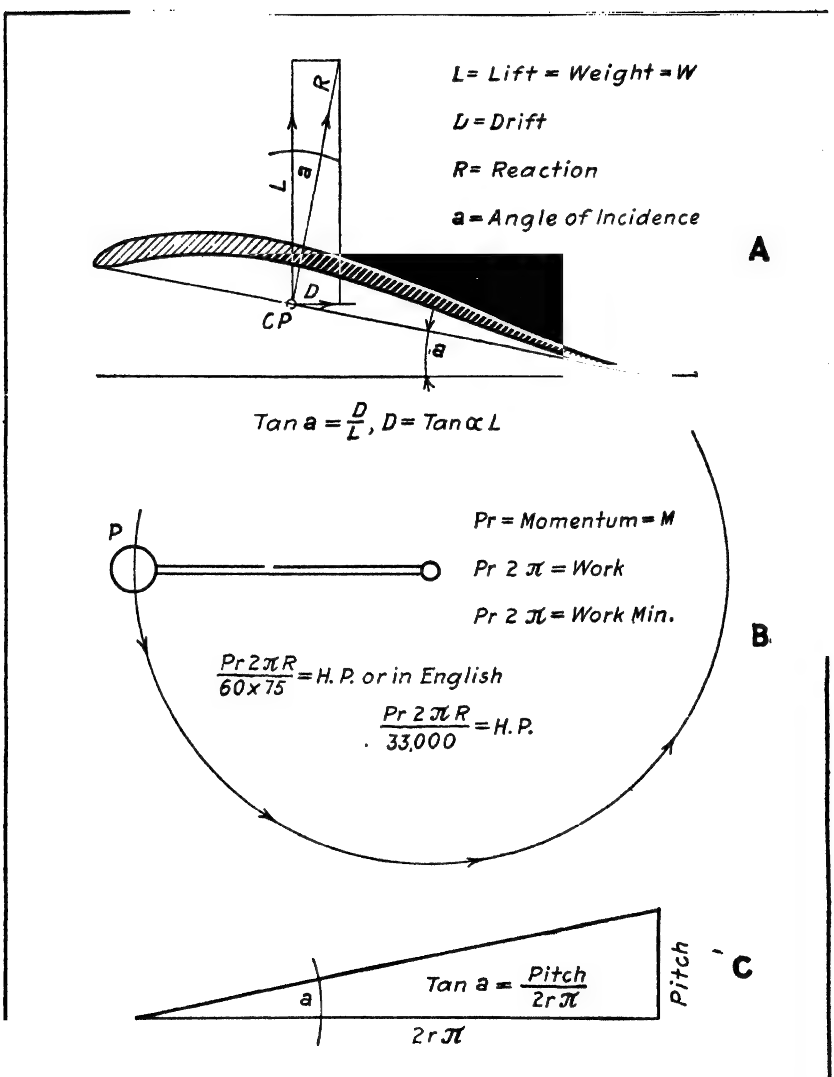


Fig. 14.—Diagrams Illustrating Computations to Determine Horsepower Required for Airplane Flights.

With the other method we shall find out the "thrust" in pounds or kilograms wanted for climbing and add it to drift and total head resistance, and we shall have the total "thrust" of our machine and we shall denote it with T, while thrust for climbing shall be T_c.

The following calculation is at our service to find out this thrust for climbing $\frac{V_c \times W}{550} = HP.$,

$$\text{thence } V_c = \frac{HP. \times 550}{W} \qquad (1)$$

$$HP. = \frac{T_c \times V}{550}, \text{ then from}$$

$$(1)\ V_c = \frac{\frac{T_c \times V}{550} \times 550}{W} = \frac{T_c \times V}{W}, \text{ thence, } T_c = \frac{V_c \times W}{V}$$

Whether T means drifts, head resistance and thrust for climbing, or drift and head resistance only, the following calculation is the same, only in the latter case, of course, we must add the horsepower required for climbing to the result to obtain the total horsepower.

Formula to Find Horsepower Needed for Flight.—Now, when we know the total thrust, we shall find the horsepower in the following manner:

We know that the $HP. = \frac{Pr\,2\,\pi\,R}{75 \times 60}$ in kilograms, or in English measure,

$$HP. = \frac{Pr\,2\,\pi\,R}{33{,}000} \text{ (Fig. 14 B)}$$

where P = pressure in klgs. or lbs.
r = radius on which P is acting.
R = Revolution/min.

When $B \times r = M$, then $HP. = \frac{M.R.\,2\,\pi}{4{,}500}$, thence,

$$M = \frac{HP. \times 4{,}500}{R\,2\,\pi} = \frac{716.2\ HP.}{R} \text{ in meter kilograms,}$$

or in English system $M = \frac{HP.\ 33{,}000}{R\,2\,\pi} = \frac{5253.1\ HP.}{R}$ in foot pounds.

Now the power on the circumference of the propeller will be reduced by its radius, so it will be $\frac{M}{r} = p$. A part of p will be used for counteracting the air and bearing friction, so that the total power on the circumference of the propeller will be $\frac{M}{r} \times \eta = p$ where η is the mechanical efficiency of the propeller. Now $\frac{\eta}{\tan \alpha} = T$, where α is taken on the tip of the propeller.

The originator of the formulæ takes α at the tip, but it can be taken, of course, at any point, but then in equation $p = \frac{M}{r}$, r must be taken only up to this point, and not the whole radius; but it is more comfortable to take it at the tip, as $\tan \alpha = \frac{\text{Pitch}}{r\, 2\pi}$ (Fig. 14 C).

Now we can write up the equation of the thrust: $T = \frac{716.2 \text{ HP.} \eta}{R\, r \tan \alpha}$, or in English measure $\frac{5253\ 1 \text{ HP.} \eta}{R\, r \tan \alpha}$ thence $\text{HP.} = \frac{T \times R \times r \tan \alpha}{716.2\, \eta}$, or in English measure $\frac{T \times R \times r \tan \alpha}{5253.1\, \eta}$.

The computations and formulæ given are of most value to the student engineer rather than matters of general interest to those interested in maintenance and repair, but are given so that a general idea may be secured of how airplane design influences power needed to secure sustained flight. It will be apparent that the resistance of an airplane depends upon numerous considerations of design which require considerable research in aerodynamics to determine accurately. It is obvious that the more resistance there is, the more power needed to fly at a given speed.

Power Used by Airplanes.—Light monoplanes have been flown with as little as six or seven horsepower for short distances, but most planes now built to carry two or three people use engines of 100 horsepower or more. Giant airplanes have been constructed having 2,000 horsepower distributed in four or more power units. The amount of power provided for an airplane of given design varies widely as many conditions govern this, but it will range from approximately one horsepower to each four or five pounds weight in the case of very light, fast machines to one horsepower to every fifteen to twenty pounds of the total weight in the case of medium speed machines. The development in airplane and powerplant design is so rapid, however, that the figures given can be considered only in the light

of general averages. Very efficient monoplanes of recent development have greatly exceeded these horsepower loadings. A small plane, the Klemm-Daimler has carried its load with but twenty horsepower and as this included two passengers, the loading was nearly 30 pounds per horsepower. A safe, empirical average figure may be taken as ten pounds per horsepower if good performance is to be secured and flights made at fairly high speed though cases are on record of planes carrying 40 and even 50 pounds per horsepower.

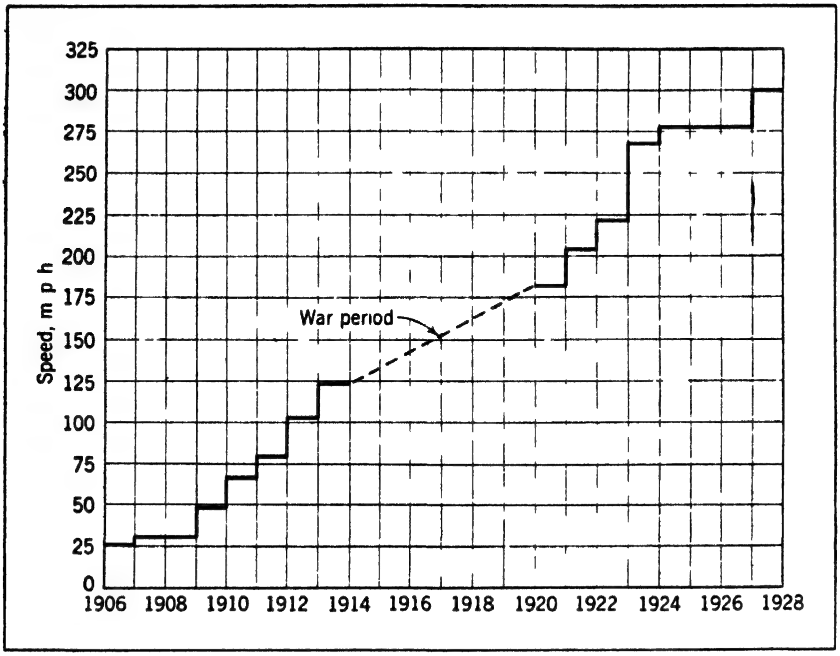


Fig. 14A.—Chart Showing Increase in Speed of Airplanes from 1906 to 1928, a Period of Twenty-two Years.

Why Explosive Motors are Best.—Internal combustion engines are best for airplanes and all types of aircraft for the same reasons that they are universally used as a source of power for automobiles. The gasoline engine is the lightest known form of prime mover and a more efficient one than a steam engine, especially in the small powers used for airplane propulsion. It has been stated that by very careful designing a steam plant and engine could be made that would be practical for airplane propulsion, but even with the latest development it is doubtful if steam power can be utilized in aircraft to as good advantage as modern gasoline-engines are. While the steam-engine is considered very much simpler than a gas-motor, the latter is much more easily mastered by the non-technical aviator and certainly requires less attention. A weight of ten pounds per horsepower is possible in a condensing steam plant but this figure is triple what is easily secured with a gas-motor which may weigh but two pounds per horsepower in the water-cooled forms and but 1.5 pounds per horsepower

in the air-cooled types. The fuel consumption is twice as great in a steam-powerplant (owing to heat losses) as would be the case in a gasoline-engine of equal power and much less weight. The Diesel engine offers interesting possibilities, but at the present time its weight-horsepower ratio is considerably higher than possible with gasoline-engines.

Wet or Dry Weights.—While on the subject of weight per horsepower, it is well to point out the difference between "wet" and "dry" weights. The latter is often used by makers of water-cooled engines to show a very low weight per horsepower, some large engines having been built that have a dry weight of but sixteen to eighteen ounces per horsepower. By the time the cooling system is taken into consideration and the water in the radiator and cylinder jackets weighed, the total or wet weight is considerably greater and will approximate that of air-cooled engines which have no "wet" weight because they do not use water, piping, water pump or radiator, and which always have a "dry" weight. When comparisons are made between differing engine types, the total weight of powerplant ready to run must be compared as a water-cooled engine is inoperative without the cooling auxiliaries and liquid. Some modern forms of air-cooled engines suitable for airplane use are shown at Fig. 15 and some recently developed and very efficient water-cooled engines for various types of aircraft are shown at Fig. 16. These engines are shown separate from their cooling auxiliaries such as radiator and piping, though water pumps are installed.

The internal-combustion engine has come seemingly like an avalanche, but it has come to stay, to take its well-deserved position among the prime movers. Its ready adaptation to road, aerial and marine service has made it a wonder of the age in the development of speed not before dreamed of as a possibility; yet in so short a time, its power for speed has taken rank on the common road against the locomotive on the rail with its century's progress. It has made aerial navigation possible and practical, it furnishes power for all marine craft from the light canoe to the transatlantic liner. It has enabled mankind to travel at speeds of over 208 miles per hour in an automobile, 318 miles per hour in an airplane and over a mile-a-minute in a motor boat. It operates the machine tools of the mechanic, tills the soil for the farmer and provides healthful recreation for thousands by furnishing an economical means of transport by land and sea. It has been a universal mechanical education for the masses, and in its present forms represents the great refinement and development made possible by the concentration of the world's master minds on the problems incidental to internal combustion engineering.

History of Engine Development.—Although the ideal principle of explosive power was conceived some two hundred years ago, at which time experiments were made with gunpowder as the explosive element, it was not until the last years of the eighteenth century that the idea took a patentable shape, and not until about 1826 (Brown's gas-vacuum engine) that a further progress was made in England by condensing the products of combustion by a jet of water, thus creating a partial vacuum.

Brown's was probably the first explosive engine that did real work. It was clumsy and unwieldy and was soon relegated to its place among the

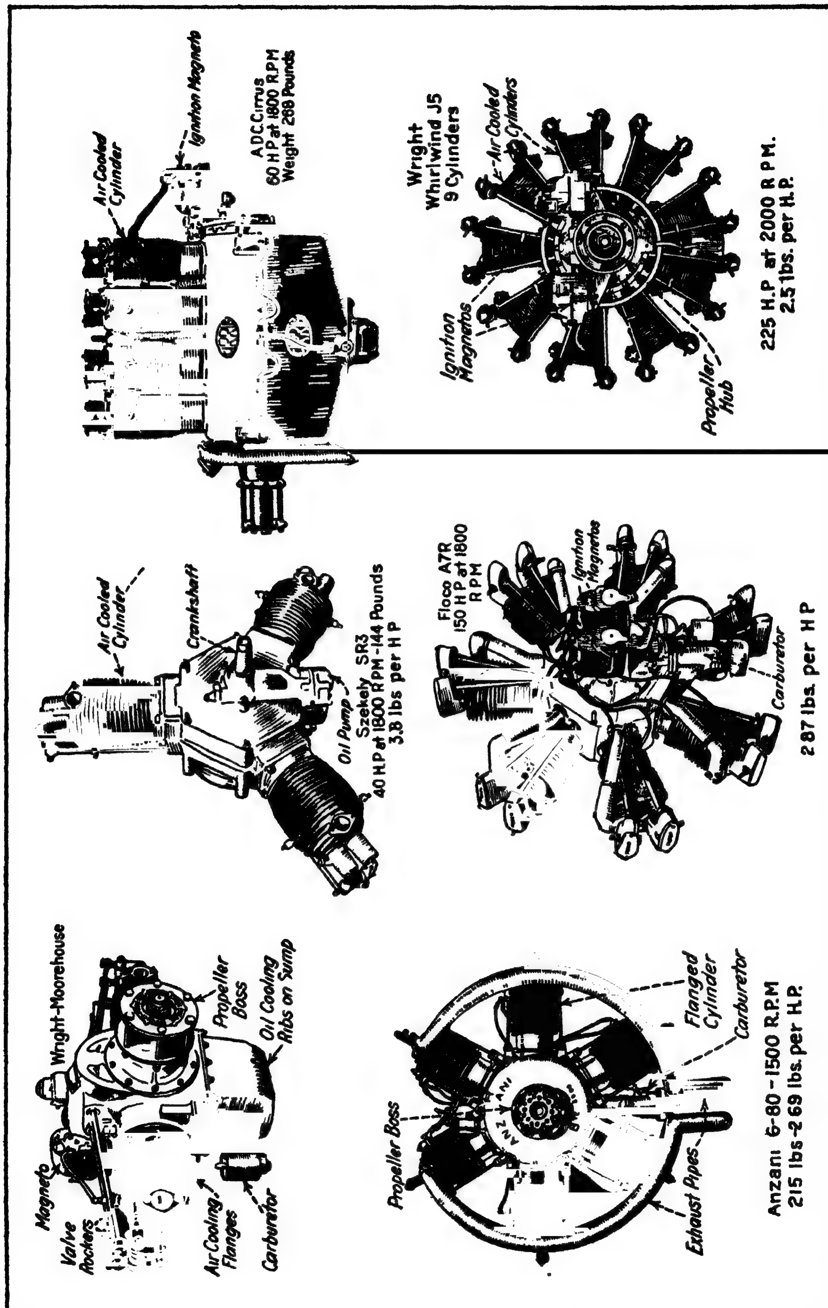


Fig. 15.—Widely Diversified Types of Air-Cooled Aviation Engines Showing Range of Design of Accepted Powerplants.

failures of previous experiments. No approach to active explosive effect in a cylinder was reached in practice, although many ingenious designs were described, until about 1838 and the following years. Barnett's engine in England was the first attempt to compress the charge before exploding. From this time on to about 1860 many patents were issued in Europe and a few in the United States for gas-engines, but the progress was slow, and its practical introduction for power came with spasmodic effect and low efficiency. From 1860 on, practical improvement seems to have been made, and the Lenoir motor was produced in France and brought to the United States. It failed to meet expectations, and was soon followed by further improvements in the Hugon motor in France (1862), followed by Beau de Rocha's four-cycle idea, which has been slowly developed through a long series of experimental trials by different inventors. In the hands of Otto and Langdon further progress was made, and numerous patents were issued in England, France, and Germany, and followed up by an increasing interest in the United States, with a few patents.

From 1870 improvements seem to have advanced at a steady rate, and largely in the valve-gear, fuel distribution, ignition and precision of governing for variable load. The early idea of the necessity of slow combustion was a great drawback in the advancement of efficiency, and the suggestion of de Rocha in 1862 did not take root as a prophetic truth until many failures and years of experience had taught the fundamental axiom that rapidity of action in both combustion and expansion is the basis of success in explosive motors.

With this truth accepted and the demand for small and safe prime movers, the manufacture of internal-combustion engines increased in Europe and America at a more rapid rate, and improvements in perfecting the details of this cheap and efficient prime mover have raised it to the dignity of a universal, standard motor and a dangerous rival of the steam-engine for small and intermediate powers in stationary applications with a prospect of largely increasing its individual units to many hundred, if not several thousand horsepower in a single cylinder. The unit size in a single cylinder has now reached to about 1,200 horsepower and by combining cylinders in the same machine, powers of from 10,000 to 20,000 horsepower are now available for large powerplants operating on the Diesel principle and applied to marine use. The use of steam-engines in the automotive field is extremely limited, the writer knowing of no practical steam powerplant suitable for airplanes.

Main Types of Internal-Combustion Engines.—This form of prime mover, especially as applied to aerial navigation, has been built in so many different types, all of which have operated with some degree of success that the diversity in form will not be generally appreciated unless some attempt is made to classify the various designs that have received practical use in automotive and stationary applications. Obviously the same type of engine is not universally applicable, because each class of work has individual peculiarities which can best be met by an engine designed with the peculiar conditions present based on duty to be performed in view.

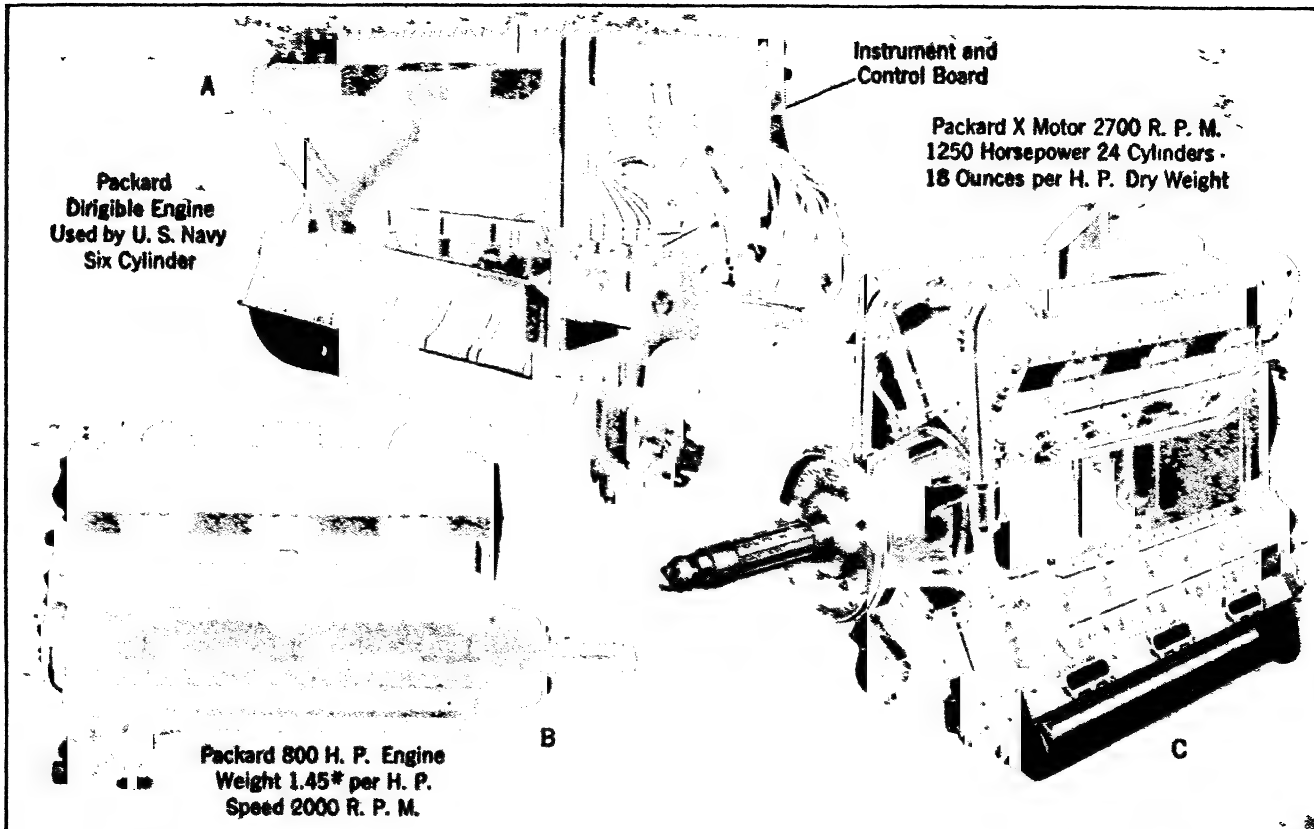


Fig. 16.—Illustration Depicting Practical Aircraft Powerplants of Packard Manufacture. A—This Type is Made for Dirigible Balloon Propulsion. That Shown at B is a Twelve Cylinder Vee Type for Airplanes. The Packard 24-Cylinder X Motor Shown at C is one of the Most Powerful Airplane Engines Yet Produced for its Weight.

The following tabular synopsis will enable the reader to judge the extent of the development of what is now the most popular prime mover for all purposes.

A. Internal Combustion (Standard Type)
   1. Single Acting (Standard Type)
   2. Double Acting (For Large Power Only)
   3. Simple (Universal Form)
   4. Compound (Rarely Used)
   5. Reciprocating Piston (Standard Type)
   6. Turbine (Revolving Rotor, not fully developed)
   7. Standard Type, Supercharged
   8. Diesel and Semi-Diesel Types

A1. Two-Stroke Cycle
   a. Two Port
   b. Three Port
   c. Combined Two and Three Port
   d. Fourth Port Accelerator
   e. Differential Piston Type
   f. Distributor Valve System
   g. Double Piston Type
   h. Diesel System
   i. Semi-Diesel System

A2. Four-Stroke Cycle
   a. Automatic Inlet Valve (Rarely Used)
   b. Mechanical Inlet Valve
   c. Poppet or Mushroom Valves
   d. Diesel Injection Type
   e. Slide Valve
      1. Sleeve Valve, Single
      2. Reciprocating Ring Valve
      3. Piston Valve
      4. Sleeve Valves, Double
      5. Combined Sleeve and Poppet Valve
   f. Rotary Valves
      1. Disc
      2. Cylinder or Barrel
      3. Single Cone
      4. Double Cone
   g. Two Piston (Balanced Explosion)
   h. Rotary Cylinder, Fixed Crank (Aerial)
   i. Fixed Cylinder, Rotary Crank (Standard Type)

A3. Six-Stroke Cycle (Rare)

B. External Combustion (Practically Obsolete)
   a. Turbine, Revolving Rotor
   b. Reciprocating Piston

Classification by Cylinder Arrangement.—Another method of classifying aviation and other automotive engines is by the number of cylinders

and their arrangement. This varies widely and many unconventional combinations of cylinders have been applied to airplanes that would not be practical in other automotive applications. Some of these follow:

Single Cylinder
- a. Vertical
- b. Horizontal
- c. Inverted Vertical

Double Cylinder
- a. Vertical
- b. Horizontal (Side by Side)
- c. Horizontal (Opposed)
- d. 45 to 90 Degrees Vee (Angularly Disposed)
- e. Horizontal Tandem (Double Acting)

Three Cylinder
- a. Vertical
- b. Horizontal
- c. Rotary (Cylinders Spaced at 120 Degrees)
- d. Radially Placed (Stationary Cylinders at 120 Degrees)
- e. One Vertical, One Each Side at an Angle (W Type)
- f. Compound (Two High Pressure, One Low Pressure)

Four Cylinder
- a. Vertical (Upright)
- b. Horizontal (Side by Side)
- c. Horizontal (Two Pairs Opposed)
- d. 45 to 90 Degrees Vee (Two Pairs Opposed)
- e. Twin Tandem (Double Acting)
- f. Fan Type
- g. X Type
- h. Vertical (Inverted)
- i. X Type (Caminez)

Five Cylinder
- a. Vertical (Five Throw Crankshaft, Unusual)
- b. Radially Spaced at 72 Degrees (Stationary)
- c. Radially Placed Above Crankshaft (Stationary) A Fan Type Motor
- d. Placed Around Rotary Crankcase (72 Degrees Spacing)
- e. Barrel type

Six Cylinder
- a. Vertical
- b. Horizontal (Three Pairs Opposed)
- c. 45 to 90 Degrees Vee
- d. Equally Spaced Radially (Two Throw Crank)
- e. Vertical, inverted

Seven Cylinder
- a. Equally Spaced (Rotary)
- b. Equally Spaced (Static Radial)
- c. Barrel type

Eight Cylinder
- a. Vertical
- b. Horizontal (Four Pairs Opposed)
- c. 45 to 90 Degrees Vee
- d. X Type (Four Pairs of Cylinders)

Nine Cylinder
- a. Equally Spaced (Rotary)
- b. Equally Spaced (Static Radial)

Ten Cylinder
- a. Static Radial (Two Five Cylinder Engines Tandem)
- b. Rotary Radial (Two Five Cylinder Engines Tandem)

Twelve Cylinder
- a. Vertical (Unusual Because of Length)
- b. Horizontal (Six Pairs Opposed)
- c. 45 to 90 Degrees Vee
- d. W Type (Three Banks of Four)
- e. X Type (Four Banks of Three)
- f. Static Radial (Six Pairs 60 Degrees Apart)

Fourteen Cylinder
- a. Rotary (Two Seven Cylinder Engines Tandem)
- b. Static Radial (Two Seven Cylinder Engines Tandem)

Sixteen Cylinder
- a. 45 to 90 Degrees Vee
- b. Horizontal (Eight Pairs Opposed)
- c. X Type (Four Banks of Four)

Eighteen Cylinder
- a. Rotary Cylinders (Two Nine Cylinder Engines Tandem)
- b. Static Radial (Two Nine Cylinder Engines Tandem)
- c. W Type (Three Banks of Six)

Twenty-Four Cylinder
- a. X Type (Four Banks of Six)

Engine Types Defined.—In order to prevent confusion, the National Advisory Committee for Aeronautics has defined various types of engines as follows:

barrel-type engine—An engine having its cylinders arranged equidistant from and parallel to the main shaft.

inverted engine—An engine having its cylinders below the crankshaft.

left-hand engine—An engine whose propeller shaft, to an observer facing the propeller from the antipropeller end of the shaft, rotates in a counterclockwise direction.

left side (engine)—That side which, to an observer looking from the antipropeller end toward the propeller end, lies on the left-hand side.

radial engine—An engine having stationary cylinders arranged radially around a common crankshaft.

right-hand engine—An engine whose propeller shaft, to an observer facing the propeller from the antipropeller end of the shaft, rotates in a clockwise direction.

right side (engine)—That side which, to an observer looking from the antipropeller end toward the propeller end, lies on the right-hand side.

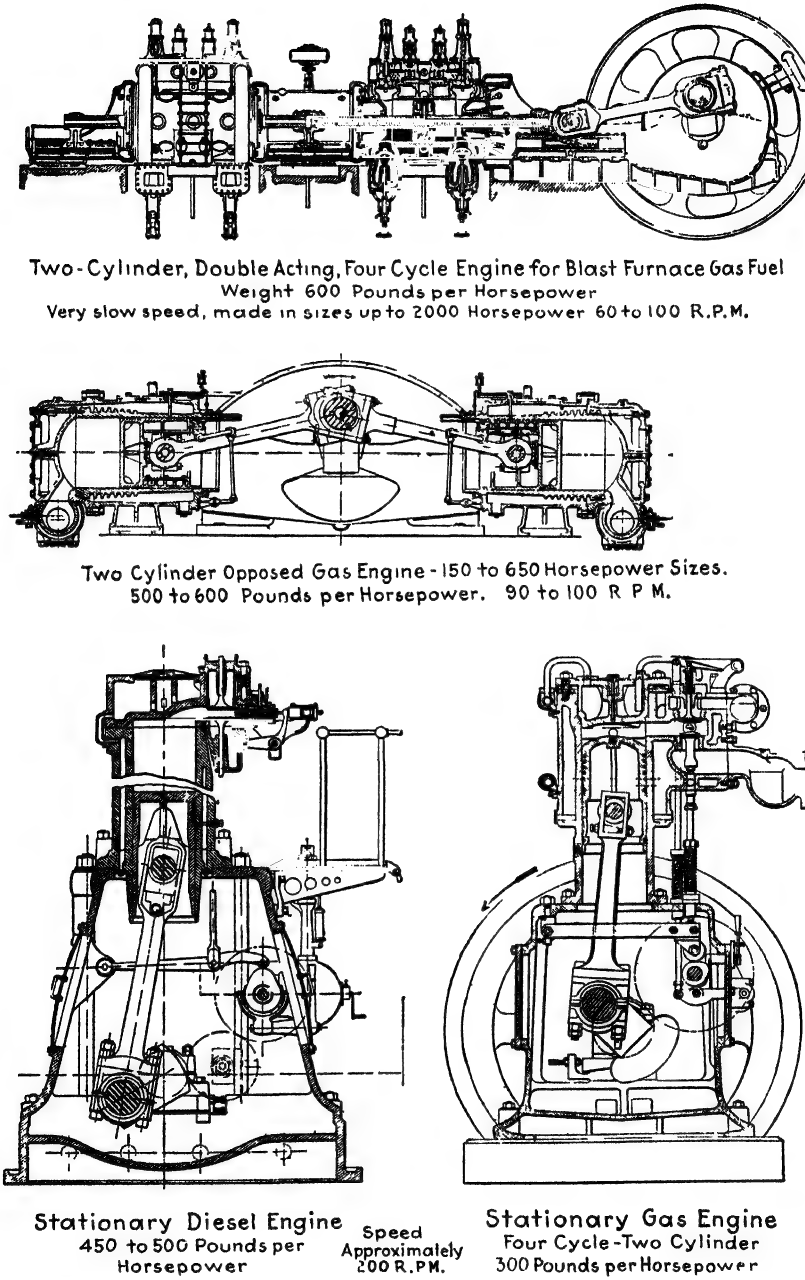


Fig. 17.—Plate Showing Heavy Slow Speed Internal-Combustion Engines Used Only for Stationary Power in Large Installations Giving Weight to Horsepower Ratio for Comparison with Aviation Engines, Showing that Ratio Augments with Decreasing Speed.

rotary engine—An engine having cylinders arranged radially with crank-case and cylinder assembly revolving around a common fixed crank-shaft.

supercharged engine—An engine with mechanical means for increasing the cylinder charge beyond that normally taken in at the existing atmospheric pressure and temperature.

vertical engine—An engine having its cylinders arranged vertically above the crankshaft.

Vee-type engine—An engine having its cylinders arranged in two rows, forming, in the end view, the letter "V."

W-type engine—An engine having its cylinders arranged in three rows, forming, in the end view the letter "W." Sometimes called the "broad-arrow type."

Of all the types enumerated above engines having less than twelve cylinders are the most popular in everything but aircraft work. The four-cylinder vertical is without doubt the most widely used of all types owing to the large number employed as automobile powerplants in large production cars such as the Ford, Chevrolet and Whippet. It has also given excellent results in aircraft work and records for reliability, endurance and economy have been made with both the D. H. Moth and the Aero Avian fitted with the A. D. C. Cirrus (English) motor. Stationary engines in small and medium powers are invariably of the single or double form. Three-cylinder engines are seldom used at the present time in marine work or in stationary forms but have been applied to some extent in Y forms to small sporting airplanes. Eight- and twelve-cylinder motors have received application practically always in automobiles, racing motor boats or in aircraft. The only example of a fourteen-cylinder motor to be used to any extent is incorporated in airplane construction. This is also true of the sixteen- and eighteen-cylinder forms and of twenty-four-cylinder X engines now in process of development.

Weight-Horsepower Ratio.—The duty an engine is designed for determines the weight per horsepower. High powered engines intended for steady service are always of the slow speed type and consequently are of very massive construction. Various forms of heavy duty type stationary engines are shown at Fig. 17. Some of these engines may weigh as much as 600 pounds per horsepower. As the crankshaft speed increases and cylinders are multiplied the engines become lighter. While the big stationary powerplants may run for years without rebuilding, airplane engines require overhauling after about 150 to 300 hours air service for the fixed cylinder types and 80 hours or less for the rotary cylinder air-cooled forms. There is evidently a decrease in durability and reliability as the weight is lessened and speed is increased. The illustrations presented in this volume as the subject is developed will permit of obtaining a good idea of the variety of forms internal-combustion engines are made in, and their installation in various types of aircraft.

Life of Aviation Engines.—Aircraft engines are usually rebuilt after 300 hours and have a life of about 1,500 hours if properly cared for. Modern high speed supercharged auto racing engines have developed over

one horsepower for each cubic inch displacement but this calls for speeds of crankshaft rotation greatly in excess of what is practical in aircraft work. Airplane engines average one horsepower for each three cubic inches displacement.

A point that should be made clear at the outset is that when average or empirical values are given, they are presented only for their value as a matter of general information and not as applying to all engines of the same general type. Some authorities, familiar with the performance of some specific type of engine might consider the values high, others, basing their opinion on another set of facts might consider the figures low.

As an example of the maintenance work done on an airplane engine of recent development, the servicing required by Colonel Lindbergh's Wright Whirlwind engine in the famous Ryan monoplane (Spirit of St. Louis) is outlined by the makers in their house organ, *The Wright Engine Builder,* during the six times it was serviced as follows:

June 13, 1927, at 85 hours—Upon Colonel Lindbergh's return from Europe the engine cam follower was found to be cracked and stuck in the guide. This was replaced with an old type cam follower from stock at the Naval Air Station, Anacostia.

July 12, 1927, at 95 hours—Engine was removed from plane, disassembled, inspected and reassembled, only replacement necessary being one exhaust tappet roller pin. Several bent push rods were straightened and rusty valve stems polished.

Nov. 3, 1927, at 355 hours—Engine was removed from plane and the following overhaul work performed:

All valves ground.

Replaced scraper ring broken in disassembly and compression ring with broken step.

Cut $\frac{1}{4}$ in. from threaded end of gasoline pump relief valve needle, recut slot and replaced battered cap.

Washed out thrust ball bearing until it spun freely.

Polished Contex idler gear pin in fuel pump.

Replaced battered Parker fitting on fuel pump.

Stoned burred compression rings on Nos. 1, 2 and 9 pistons.

Replaced ignition wires with new Airtite cable.

Jan. 12, 1928, at 415 hours—Obsolete exhaust valves replaced with latest type in Panama.

Feb. 9, 1928, at 471 hours—Engine inspected at Havana. Reground one leaking exhaust valve and checked clearances.

Feb. 13, 1928, at 485 hours—Two cracked sparkplugs replaced.

In comparing airplane and automobile engines, the latter often need an overhauling after 20,000 miles travel and seldom run more than 80,000 miles as that figure represents from four to six years normal service. The automobile engine is operating at less than capacity over 90 per cent of the time it is in use, most of the time it is being run at about 25 per cent of its power. An airplane engine is usually operated in the range from 75 to 100 per cent of its power. It is so highly refined at the present day that it may run as many miles at open throttle and full power output as an automobile engine does at quarter throttle. Air-cooled aviation engines

have run 300 hours without an overhaul, which corresponds to 30,000 miles service if we assume a speed of 100 miles per hour. There is no stock automobile engine built that could run even half of this mileage at wide open throttle without requiring overhauling. After overhauling, an airplane engine returns to service and some modern engines are capable of undergoing four overhauls and running for periods ranging from 800 to 1,500 hours without wearing out, or 80,000 to 150,000 miles service at practically its full power output or at least, three-quarters of its full capacity.

Future Engine Development.—Turning now to possible future developments it may be of interest to speculate toward what direction progress in aircraft engines will lead. Problems to be solved in this field are well known and consist largely of detailed improvements intended to yield lighter and more reliable engines that will be more economical with respect both to first cost and to operation and maintenance. Although much experimental effort in engine development is being continually directed along unconventional lines, such as the barrel and cam types, and engines employing the Diesel or semi-Deisel cycle, it is reasonable to believe that during the next few years important advances will be made by conventional twelve-cylinder water-cooled engines and by nine-cylinder fixed-radial air-cooled engines, the two types that offer the best possibilities for immediate engineering advance and where very high powers are required from a single motor the X form of 24 cylinders, the W form of eighteen cylinders and the double banked nine-cylinder or eighteen-cylinder radial air-cooled engines are all possibilities that have already been realized in experimental types.

It is reasonable to look forward to having available in the near future engines that will weigh about one pound per horsepower; and, concurrently with this development, considerable effort will undoubtedly be devoted toward reducing the specific fuel-consumption. For it should be borne in mind that an engine weighing one pound per horsepower will, at the present rate of fuel-consumption, consume its own weight of fuel every two hours. Doped fuels and higher compressions will make possible higher power outputs for a given cylinder displacement. Ethyl-gas is said to permit the use of pressures up to 150 pounds per square inch before ignition without risk of premature ignition or detonation after ignition. The compression limit with aviation gasoline is about 110 to 120 pounds, so a 25 per cent increase in mean effective pressure is possible without changing engine designs other than augmenting the compression and increasing strength of parts to withstand augmented stresses produced by increased explosion pressures.

A marked increase in the brake mean effective pressure of aircraft engines during the war period was due principally to improvement in sparkplugs. The compression-ratio is limited by the detonation characteristics of the available fuel; the highest ratio that is regarded as advisable when using domestic aviation-gasoline is 5.5 to 1.0. No radical increase in engine-power with present designs is to be expected except as the result of raising the normal engine-speed or of the use of nondetonating fuel. By the use of superchargers it is expected that the mean effective pressure can be kept at about 140 pounds per square inch at the higher engine-speeds, which will mean a tremendous output of power from a small engine. However,

the effect of the use of tetra-ethyl lead in fuel in permitting higher compressions as well as the use of a supercharger offers interesting possibilities for future development as this will reduce fuel-consumption from the present figure by greatly increasing the power output possible from a given cubic capacity of the engine, which means an increase in volumetric efficiency. The lightest engine yet made, namely the 900-hp. Bristol Mercury aeronautic engine, weighs exactly eleven ounces per horsepower, complete with all its auxiliary gear, while the average weight of radial aeronautic engines today is 1.4 lb. per hp. or just half of what it was ten years ago. The normal maximum output of a commercial engine twenty years ago was approximately six horsepower per liter of cylinder capacity; ten years ago it had risen to twelve horsepower; and today it is well over twenty horsepower; yet to all outward appearances the engine of today differs but little from that of twenty years ago. In this short period it has been learned how, by purely detail design, to increase the efficiency by more than 50 per cent and the speed by more than 120 per cent.

Airplane Engine Costs.—The reader who has priced airplane engines which are offered by the builders may think such engines are greatly overpriced if he bases his conclusions on cost of automobiles, for example. Airplane engine output is limited because the demand is limited and much of the high cost can be attributed to the high cost of production inevitable when engines are produced in small quantities. As the demand increases and the production augments the prices will become lower. Airplane engines, especially the types built for military purposes will never be cheap because of the nature of the materials used and the expensive machining and inspection processes will make a relatively high cost imperative, no matter how many engines are built. Commercial engines, however, will become cheaper as output increases.

Figures taken by considering various forms of stationary gasoline engines show a price range of from $10 to $20 per horsepower. Diesel and kerosene burning engines may cost as much as $60 per horsepower. A good steam-engine costs $25 to $30 per horsepower for the engine alone and it is useless without expensive auxiliaries such as boilers, pumps, condensers, etc., that will bring the cost of the complete installation much higher. Motor boat engines sell for from $20 to $30 per horsepower. Outboard motors may cost as high as $70 per horsepower for a small engine to $35 per horsepower for a five to six horsepower size. Automobile engines cost from $10 to $40 per horsepower when purchased separately and are not capable of maintaining their full power output nearly as long without trouble as even the poorest of aviation engines. Few stock or even racing automobile engines could go through the 50 hour test at full throttle that airplane engines pass and show as little wear and tear.

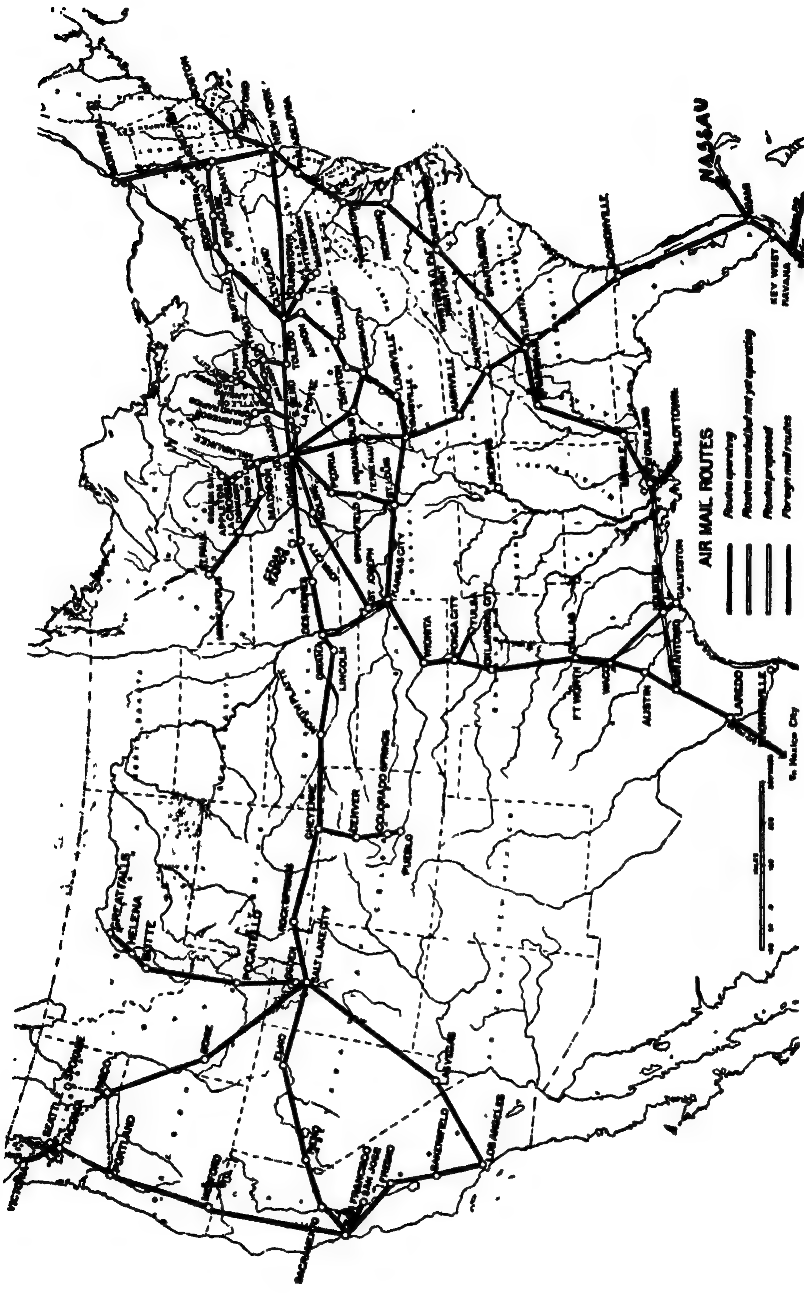
The writer was recently quoted a price of $1,200 on an aviation engine rated at 80 horsepower at 2,200 r.p.m., and this was not a wartime left-over but a new, recently developed small radial air-cooled engine. This brings the cost to $15.00 per horsepower. An analysis of costs of aviation engines by an aeronautical magazine (*Aviation*) made recently gave an average value of $20 per horsepower for aircraft engines so if one compares the

cost with that of other engines in performance or endurance, one will concede that such engines are not unduly expensive even at their present prices.

Air Mail Shows Aviation's Possibilities.—On the twenty-fifth anniversary of the first American railroad, there were only 9,051 miles of railway tracks in the United States. If the comparison with the 17,170 miles of Civil Airways in this country on the twenty-fifth anniversary of the first flight may be taken as an index, air transportation will serve every city in the country before many years. In opening up new territory for development the airplane has a tremendous advantage over any other form of transportation. All that it needs are terminal ports with occasional intermediate emergency fields. The railroad must not only have these facilities, but they must be connected by road beds and steel rails. Even motor bus and truck transportation requires good roads for efficient operation. Railroads and highways are expensive to build. Mountainous country or large areas of low swamp land are serious difficulties from both engineering and financial points of view. The highway of the air is as free to man as it is to the birds. Already the airplane is playing an important part in developing the rich natural resources of remote parts of Canada and Alaska, Central and South America, Asia, Africa, and Australia.

Clothed in various forms, the statement has been made repeatedly that the United States Air Mail is the greatest co-ordinated Air Transport system in the world. As the year 1928 draws to a close this statement is substantiated by figures that are incontrovertible. On December 31, 1927, the Air Mail routes of the United States covered 8,450 miles, over which daily flights were made totalling 18,748 miles. During 1928 new Air Mail routes were put in operation covering 6,176 miles, or 73 per cent of all the mileage previously operated. Over these routes mail is flown 12,352 miles daily. By adding the totals of mileage inaugurated in 1928 to that of the routes previously in operation, we find that at this writing the United States Air Mail routes cover 14,626 miles of Civil Airways, flying 31,352 miles a day.

Night Flying Over Lighted Airways.—Perhaps more noteworthy than any other particular feature is the great development of night flying over lighted airways. It is this overnight service between cities a thousand miles apart that is making Air Mail of such tremendous value to business and banking and private individuals. For example, the Atlanta-New York Air Mail route is one of the lines flown entirely at night. The two cities are 763 miles apart on an air line, but a letter mailed at 6 p. m. in Atlanta is delivered in New York in the same delivery as a letter mailed in Newark—fifteen miles away—at the same hour. Of the 14,626 miles of Civil Airways 9,341 miles or 64 per cent are lighted for night flying. The figures presented in the preceding paragraphs, it will be noted, refer only to Air Mail routes. In addition to the mail operations there are numerous passenger and express lines flying regular daily schedules between American cities. There are 2,544 miles of these routes, flying a total of 12,902 miles a day. This make a grand total for all mail, express and passenger air lines of 17,170 miles of airways over which planes fly on regular schedules 44,254 miles per day.



14,626 Miles of Airmail Routes, December 31, 1928.

It is frequently said that the United States, the birth place of heavier-than-air flight, lags disgracefully behind Europe in Air Transportation development. This was unfortunately true until this year, but at this writing we can justly let figures speak for themselves. In 1927 on all European Airways 12,616,752 miles were flown. An extremely conservative estimate for mileage flown in 1928 on scheduled passenger, mail or express flights in the United States is 13,500,000 miles. This estimate is based on only 300 days of the year. In addition to this, the Department of Commerce estimates that over 30,000,000 miles were flown in 1928 in the United States by air service operators on unscheduled flights.

True, this country has lagged behind Europe in passenger Air Transport, but this situation has been considerably alleviated in 1928. At this time 42 American air routes are flying passengers on regular daily schedules. Many of the large companies now operating mail or express routes are definitely planning to inaugurate passenger service in the near future. Some companies, such as Transcontinental Air Transport and Pitcairn Aviation, Inc., have announced passenger Air lines by day that will connect with railroads over which the passengers may continue their journey by night.

In recording this account of Air Transportation development in the United States in 1928. It is fitting to comment on the one outstanding event which contributed more to the development of Air Mail than any other feature. This was the inauguration of the five cent Air Mail rate by the Post Office Department on August 1st, 1928. The immediate effect of this action was to approximately double the amount of Air Mail flown, from 200,000 pounds in July to 400,000 pounds in August. Since then the poundage has steadily increased, with every indication that it will exceed half a million pounds in December. This is as much Air Mail as was carried in any three months prior to August of this year.

Future Airplanes and Commercial Possibilities.—With this growing support by the government and the people of the United States, it is inconceivable that Air Transportation in this country will not continue to rapidly accelerate its growth. No one can say what the future holds in store in airial transportation. At this writing the plans have been made public for a tremendous flying boat designed by Doctor Rumpler of Berlin. It is in effect a "flying wing," with ten motors of a thousand horsepower each, in the trailing edge of the wing, which also accommodates the passengers and baggage. This is a radical departure from established practice, in that the wing itself carries all of the useful load and pay load, the fuselage serving only as the support of the steering airfoils and the landing supports. Perhaps this is an indication of the future airplane, just as the doubling of Air Mail volume since August 1st is an indication of the day when Air Mail will be reckoned in tons instead of pounds. As far as sustained flight is concerned, the U. S. Army Fokker-Whirlwind monoplane "Question Mark" under command of Major Carl Spatz broke all world's records for time in the air by flying 150 hours and 40 minutes over California, the flight starting on January 1st, 1929, and lasting six and one-quarter days. This record was made possible by refueling the airplane and transferring other supplies from "tender" airplanes while in flight. This also

opens up vast commercial possibilities as well, as nonstop transcontinental flights with a pay load now seem possible.

QUESTIONS FOR REVIEW

1. Into what two branches is the science of Aeronautics divided?
2. Name principal types of heavier-than-air aircraft.
3. Name principal types of lighter-than-air aircraft.
4. What is the difference between a helicopter and an airplane?
5. Why do some airplanes use more than one motor?
6. What is the limiting factor to engine size?
7. Why must aviation engines be light?
8. What is the effect of resistance on power required to fly?
9. How much power do airplanes need?
10. What is the difference between "wet" and "dry" weights?
11. Name main types of aviation engines.
12. Define—"barrel type engine"; "rotary engine"; "supercharged engine."

CHAPTER II

OPERATING PRINCIPLES OF TWO- AND FOUR-STROKE ENGINES—ELEMENTARY THERMODYNAMICS

Four-Cycle Action—Two-Cycle Action—Comparing Two-Cycle and Four-Cycle Types—Charge Distribution—Double Piston Supercharged Two-Cycle Engines—What is Work?—Heat is a Form of Energy—Measuring Intensity of Heat—Measuring Amount of Heat—Meaning of Specific Heat—First Law of Thermodynamics—Relation of Heat and Work—Laws of Gases—Meaning of Absolute Scale—Fundamentals of Thermodynamics—Specific Heat of Gases at Constant Volume—Specific Heat of Gases at Constant Pressure—Constant Pressure Expansion—Constant Temperature or Isothermal Expansion—Adiabatic Expansion—Actual Expansion Curves—Graphic Representation of Fuel Efficiency.

Before discussing the construction of the various forms of internal-combustion engines it may be well to describe the operating cycle of the types most generally used. The two-cycle engine is the simplest because there are no valves in connection with the cylinder, as the gas is introduced into that member and expelled from it through ports cored into the cylinder walls. These are covered by the piston at a certain portion of its travel and uncovered at other parts of its stroke. In the four-cycle engine the explosive gas is admitted to the cylinder through a port at the head end closed by a valve, while the exhaust gas is expelled through another port controlled in a similar manner. These valves are operated by mechanism distinct from the piston.

Four-Cycle Action.—The action of the four-cycle type may be easily understood if one refers to illustrations at Figs. 18 and 19. It is called the "four-stroke engine" because the piston must make four strokes in the cylinder for each explosion or power impulse obtained. The principle of the gas-engine of the internal combustion type is similar to that of a gun, i.e., power is obtained by the rapid combustion of some explosive or quick burning substance. The bullet is driven out of the gun barrel by the pressure of the gas evolved by the rapid heating and inevitable expansion when the charge of powder is ignited. The piston or movable element of the gas-engine is driven from the closed or head end to the crank end of the cylinder by a similar expansion of gases resulting from combustion. The first operation in firing a gun or securing an explosion in the cylinder of the gas-engine is to fill the combustion space with combustible material. This is done in the engine by a down stroke of the piston during which time the inlet valve opens to admit the gaseous charge forced in by air pressure to the cylinder interior. This operation is shown at Fig. 18 A. The second operation is to compress this gas which is done by an upward stroke of the piston as shown at Fig. 18 B. When the top of the compression stroke is reached, the gas is ignited and the piston is driven down toward the open end of the cylinder, as indicated at Fig. 19 C. The fourth operation or exhaust stroke is performed by the return upward movement of the piston as shown at Fig. 19 D during which time the exhaust valve is opened to permit the burnt gases to leave the cylinder. As soon as the

piston reaches the top of its exhaust stroke, the energy stored in the fly-wheel rim during the power stroke causes that member to continue revolving and as the piston again travels on its down stroke the inlet valve opens and admits a charge of fresh gas and the cycle of operations is repeated.

The illustrations at Fig. 20 show how the various cycle functions take place in an L head type water-cooled cylinder engine. The sections at A and C are taken through the inlet valve, those at B and D are taken through the exhaust valve as the valves are side by side in the chamber

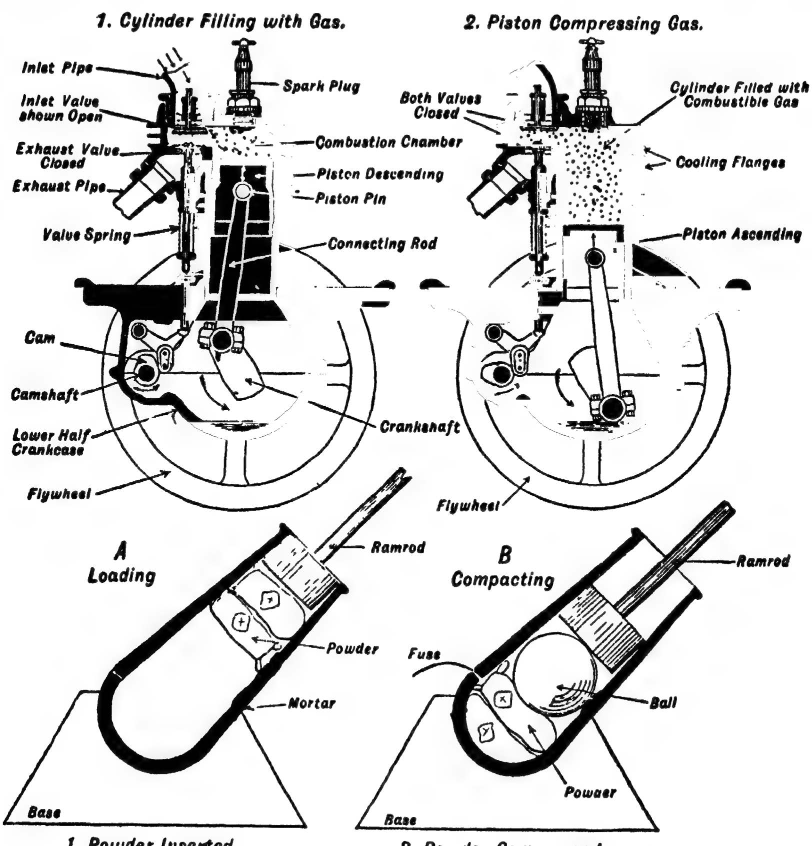


Fig. 18.—Diagrams Explaining Action of First Two Strokes of Piston in a Four-Cycle Engine, in Comparing them to Similar Functions when Loading a Cannon.

extending from the side of the cylinder, so one valve hides the other in a sectional view. This operation of valves can be studied to better advantage by the novice by consulting Fig. 21 which shows the overhead valve construction usually followed by designers of aviation engines and has the camshafts separated, i.e. showing one for operating the inlet valves and one for actuating the exhaust valves, whereas in practice, usually but one shaft carrying both sets of cams is used to operate both intake and exhaust

valves. Some engines have two overhead shafts, side by side, one for each set of valves as will be shown later.

Two-Cycle Action.—The two-cycle engine works on a different principle, as while only the combustion chamber end of the piston is employed to do useful work in the four-cycle engine, both upper and lower portions are called upon to perform the functions necessary to two-cycle engine operation. Instead of the gas being admitted into the cylinder as is the case with the four-stroke engine, it is first drawn into the engine base where it

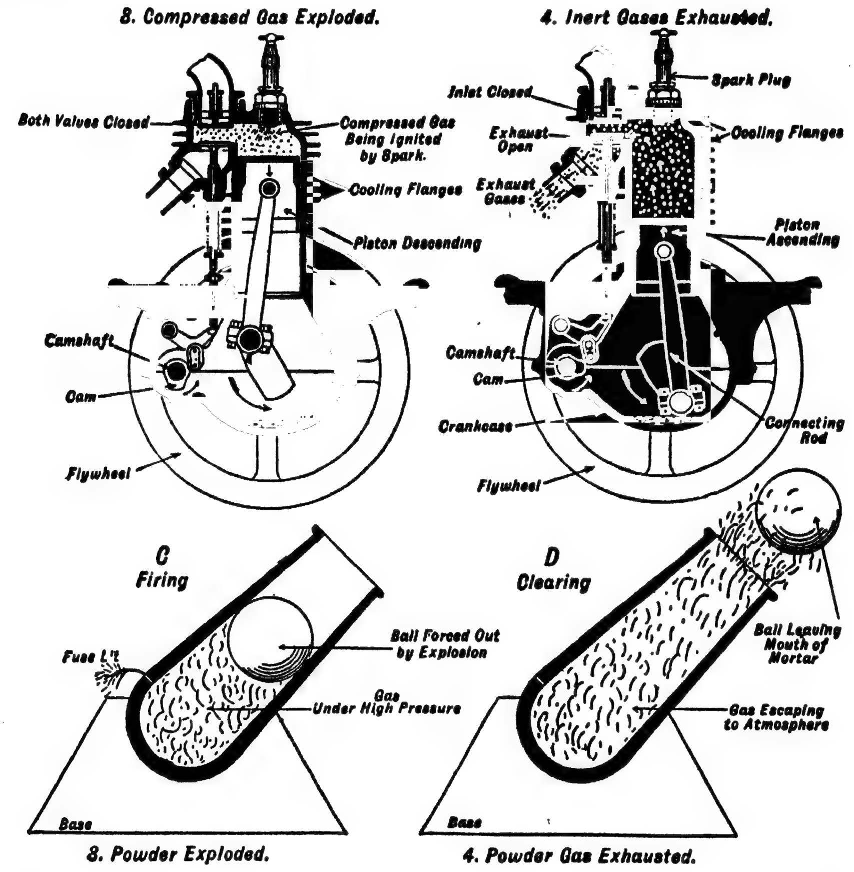


Fig. 19.—Diagrams Outlining the Second Two Strokes of the Piston in a Four-Cycle Engine, Given in Comparison with the Explosion of Powder and Discharge of Shot in a Cannon.

receives a preliminary compression prior to its transfer to the working end of the cylinder. The views at Fig. 22 should indicate clearly the operation of the two-port two-cycle engine. At A the piston is seen reaching the top of its stroke and the gas above the piston is being compressed ready for ignition, while the suction in the engine base causes the automatic valve to open and admits mixture from the carburetor to the crank case. When the

piston reaches the top of its stroke, the compressed gas is ignited and the piston is driven down on the power stroke, compressing the gas in the engine base.

When the top of the piston uncovers the exhaust port the flaming gas

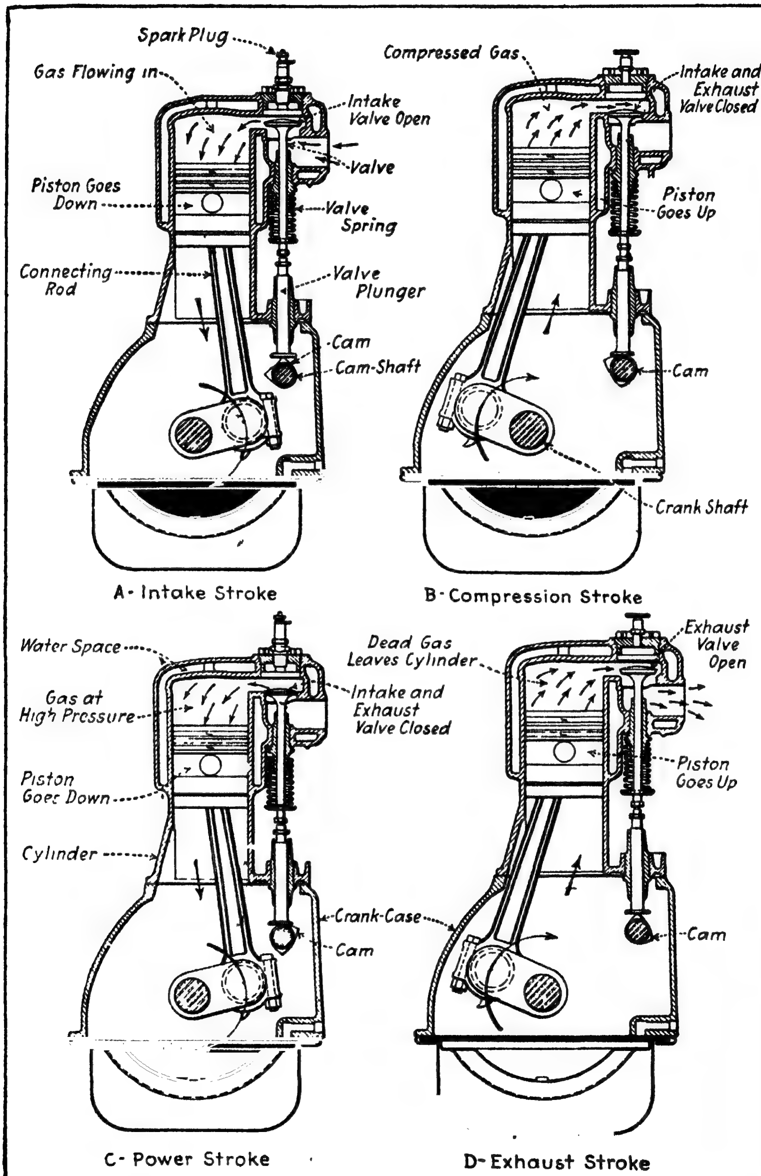


Fig. 20.—Sectional Views of L Head Gasoline-Engine Cylinder Showing Piston Movements During Four-Stroke Cycle.

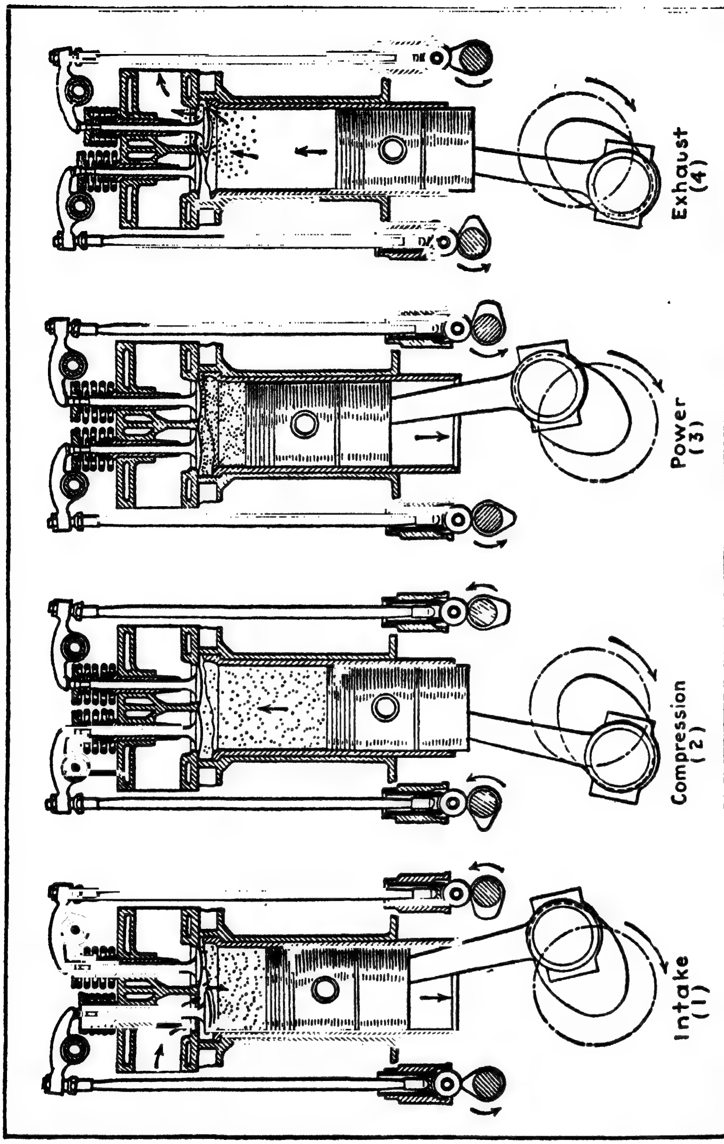


Fig. 21.—Diagram Showing Sequence of Cycles in an Internal-Combustion Engine. Gas Entering on the Intake Stroke 1, is Compressed on the Next Upward Stroke of the Piston 2, is Exploded at the End of Compression Stroke 3, and Drives the Piston Down, Imparting Power to the Crankshaft, as at 4. The Gases are then Expelled from Cylinder by the Return Movement of Piston Through the Open Exhaust Valve, Opened by the Cam Actuated Mechanism.

escapes because of its pressure. A downward movement of the piston uncovers the inlet port opposite the exhaust and permits the fresh gas to bypass through the transfer passage from the engine base to the cylinder. The conditions with the intake and exhaust port fully opened are clearly shown at Fig. 22 C. The deflector plate on the top of the piston directs the entering fresh gas to the top of the cylinder and prevents the main portion of the gas stream from flowing out through the open exhaust port. On the next upstroke of the piston the gas in the cylinder is compressed and the inlet valve opened, as shown at A to permit a fresh charge to enter the engine base.

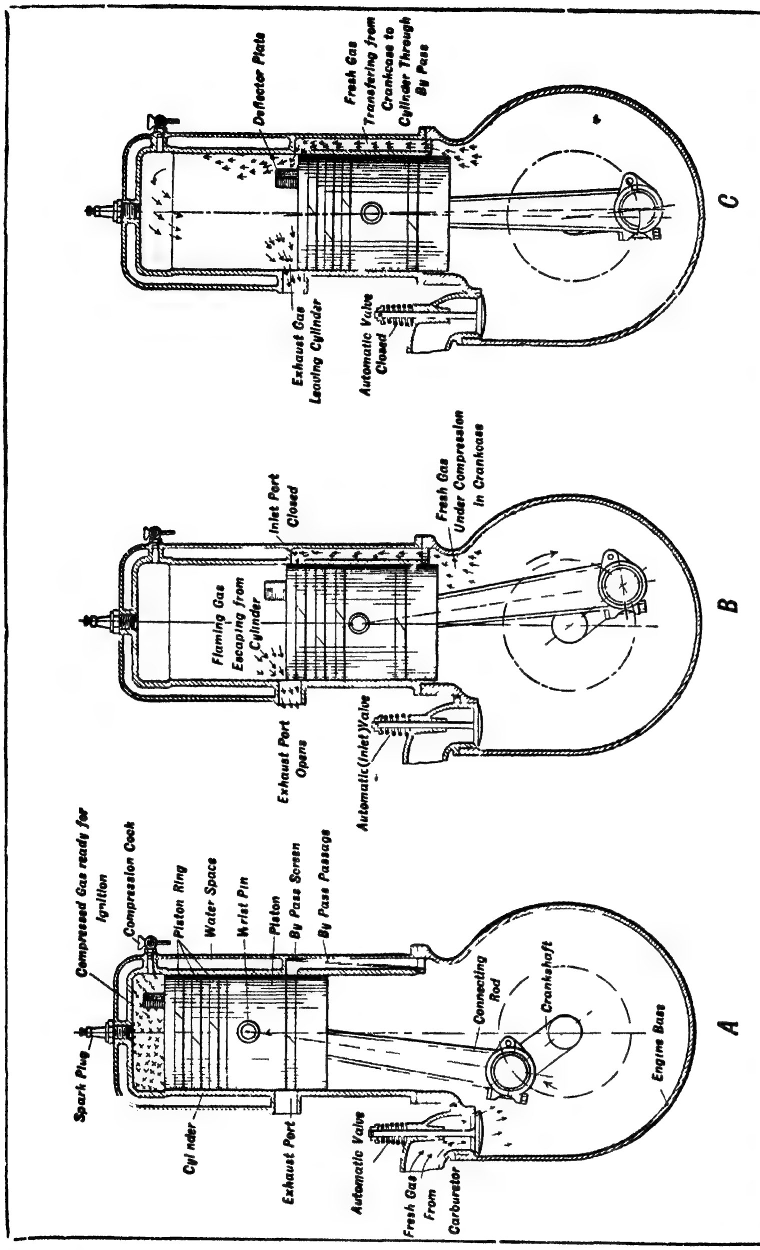


Fig. 22.—Diagrams Showing Two-Port, Two-Cycle Engine Operation. A—Compressed Gas Ready for Ignition in Cylinder Above the Piston, Fresh Gas Entering Crankcase. B—End of Power Stroke, Exhaust Ports Open and Fresh Gas in Crankcase is Transferred to Cylinder. C—Conclusion of Exhaust Stroke, Fresh Gas Transferred from Crankcase to Cylinder Through Bypass, Forcing Out Part of Burned Gas.

The operating principle of the three-port, two-cycle engine is practically the same as that previously described with the exception that the gas is admitted to the crankcase through a third port in the cylinder wall, which is uncovered by the piston when that member reaches the end of its upstroke. The action of the three-port form can be readily ascertained by studying the diagrams given at Fig. 22. Combination two- and three-port

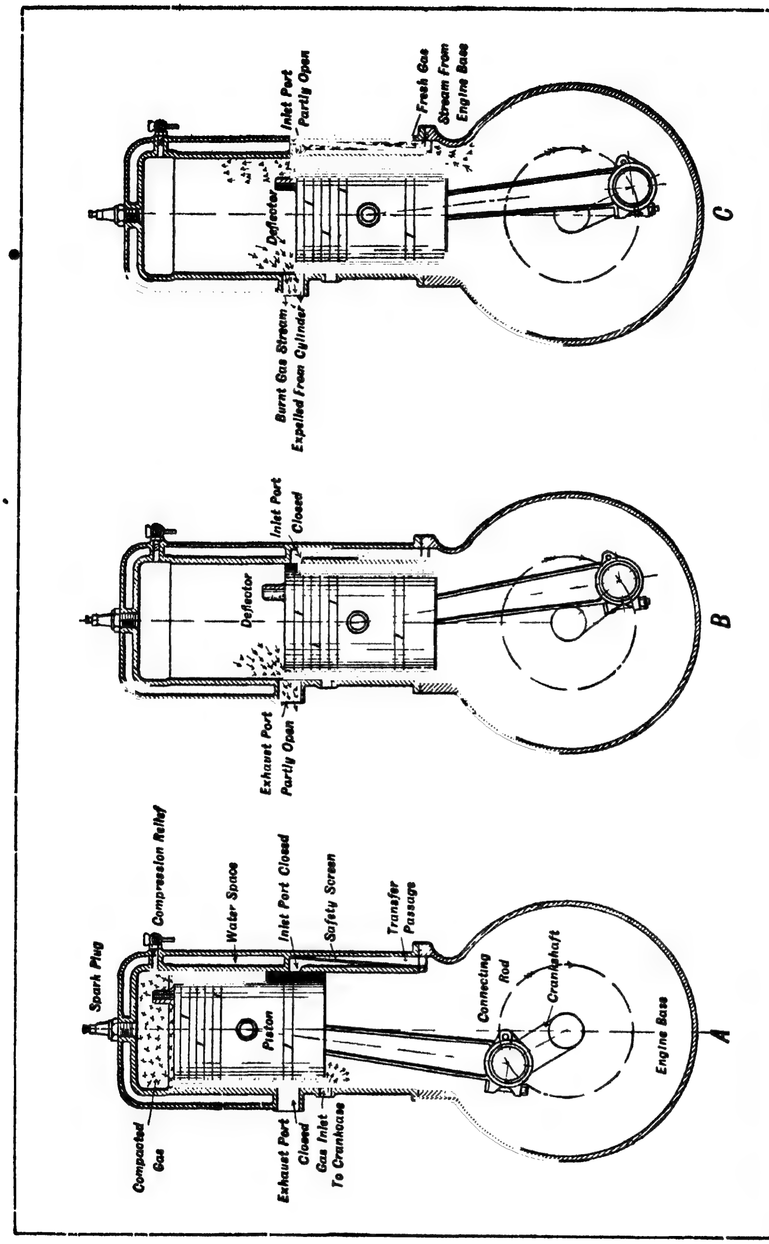


Fig. 23.—Illustration Defining Three-Port, Two-Cycle Engine Action. A—Piston Approximately at Firing Point and Uncovers Port Allowing Gas Inlet to Crankcase. B—Exhaust Port Starts to Open, Inlet Transfer Port Closed. C—Exhaust Port Nearly Fully Open and Inlet Port Uncovered by Piston to Admit Fresh Gas From the Engine Base.

engines have been evolved and other modifications made to improve the action.

Comparing Two-Cycle and Four-Cycle Types.—In the earlier years of explosive-motor progress was evolved the two types of motors in regard to the cycles of their operation. The early attempts to perfect the two-cycle principle were for many years held in abeyance from the pressure of interests in the four-cycle type, until its simplicity and power possibilities were

demonstrated by Mr. Dugald Clerk in England, who gave the principles of the two-cycle motor a broad bearing leading to immediate improvements in design, which has made further progress in the United States, until at the present time it has an equal standard value as a motor-power in some applications as its ancient rival the four-cycle or Otto type, as demonstrated by Beau de Rocha in 1862.

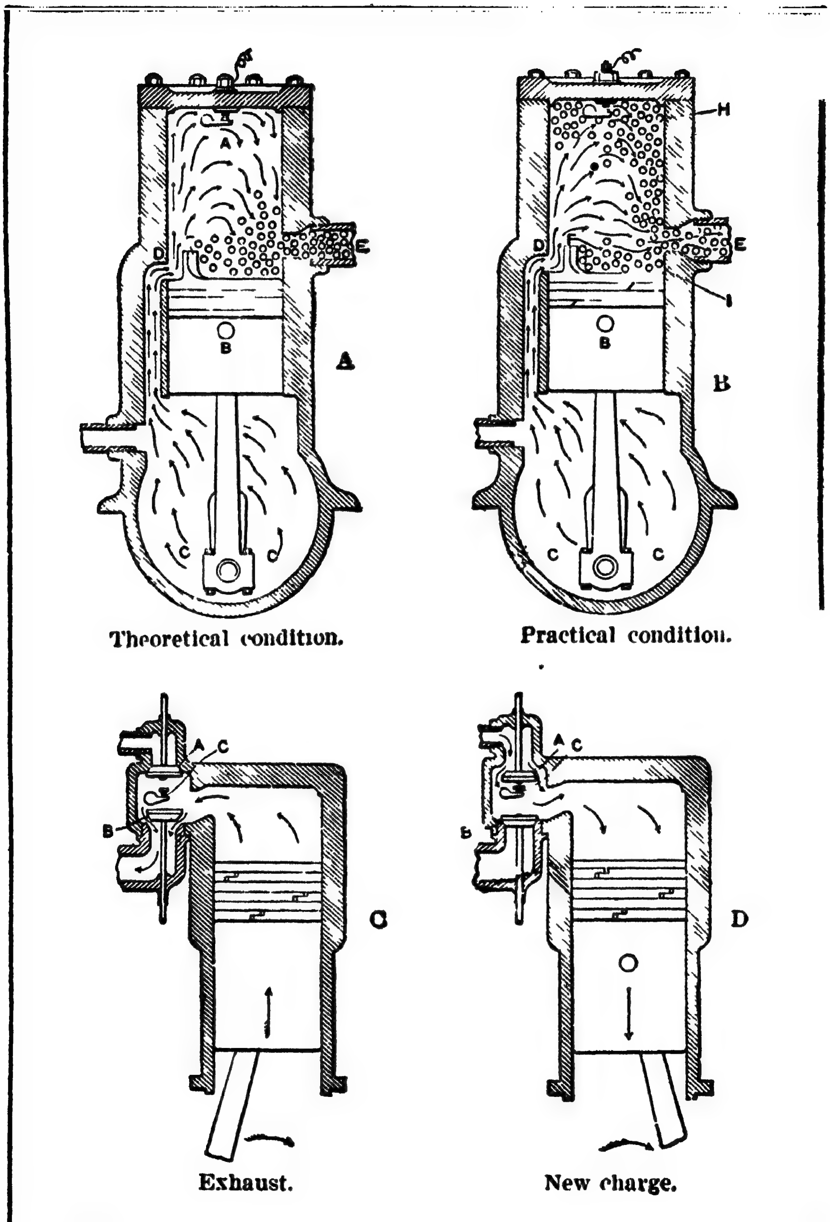
Thermodynamically, the methods of the two types are equal as far as combustion heat is concerned, and compression does favor in large degree the four-cycle type as well as the purity of the charge. The cylinder volume of the two-cycle motor is much smaller per unit of power as the explosions occur twice as often as they do in a four-stroke cylinder and the enveloping cylinder surface is therefore greater per unit of volume. Hence more heat is carried off by the jacket water during compression, and the higher compression available from this tends to increase the economy during compression which is lost during expansion to some extent.

From the above considerations it may be safely stated that a *lower* temperature and higher pressure of charge at the beginning of compression is obtained in the two-cycle motor, greater weight of charge and greater specific power of higher compression resulting in higher thermal efficiency. The smaller cylinder for the same power of the two-cycle motor gives less friction surface per impulse than of the other type; although the crank-chamber pressure may, in a measure and often does balance the friction of the four-cycle type. Probably the strongest points in favor of the simple two-cycle type are the lighter construction and the absence of valves and valve gear, making this type the most simple in construction and the lightest in weight for its developed power. Yet, for the larger power units, the four-cycle type will no doubt always maintain the standard for efficiency and durability of action. The theoretical superiority of the two-cycle is not always borne out by practical considerations because it does not give twice the power of a four-cycle of equivalent piston displacement and r. p. m., even though it gives twice as many impulses per revolution of the crankshaft. The two-cycle is not as economical in fuel consumption per horsepower developed as the four-cycle type is. Though it has twice the power strokes per cylinder, the output is increased only about 1.5 times instead of two times as a hasty consideration might indicate.

Charge Distribution.—The distribution of the charge and its degree of mixture with the remains of the previous explosion in the clearance space, has been a matter of discussion for both types of explosive motors. In Fig. 24 A we illustrate what theory suggests as to the distribution of the fresh charge in a two-cycle motor, and in Fig. 24 B what is the probable distribution of the mixture when the piston starts on its compressive stroke. The arrows show the probable direction of flow of the fresh charge and burnt gases at the crucial moment.

In Fig. 24 C is shown the complete out-sweep of the products of combustion for the full extent of the piston stroke of a four-cycle motor, leaving only the volume of the clearance to mix with the new charge and at D the manner by which the new charge sweeps by the ignition device, keeping it cool and avoiding possibilities of pre-ignition by undue heating of the terminals of the sparking device. Thus, by enveloping the sparking

device with the pure mixture, ignition spreads through the charge with its greatest possible velocity, a most desirable condition in high-speed motors with side-valve chambers and sparkplugs within the valve chamber, a common method of constructing automobile engines.



Theoretical condition.

Practical condition.

Exhaust.

New charge.

Fig. 24.—Diagram Contrasting Action of Two- and Four-Cycle Cylinders on Exhaust and Intake Strokes to Show Superior Scavenging of Four-Cycle Cylinder.

Double Piston Supercharged Two-Cycle Engines.—The development of the supercharger and its application to the four-cycle type of engine has resulted in the design of a double piston type of two-cycle engine in which an effort is made to secure the introduction of a greater weight of charge. This development has been carried on in Germany by Junkers, and in France by Causan. The F. I. A. T. interests in Italy are also working on

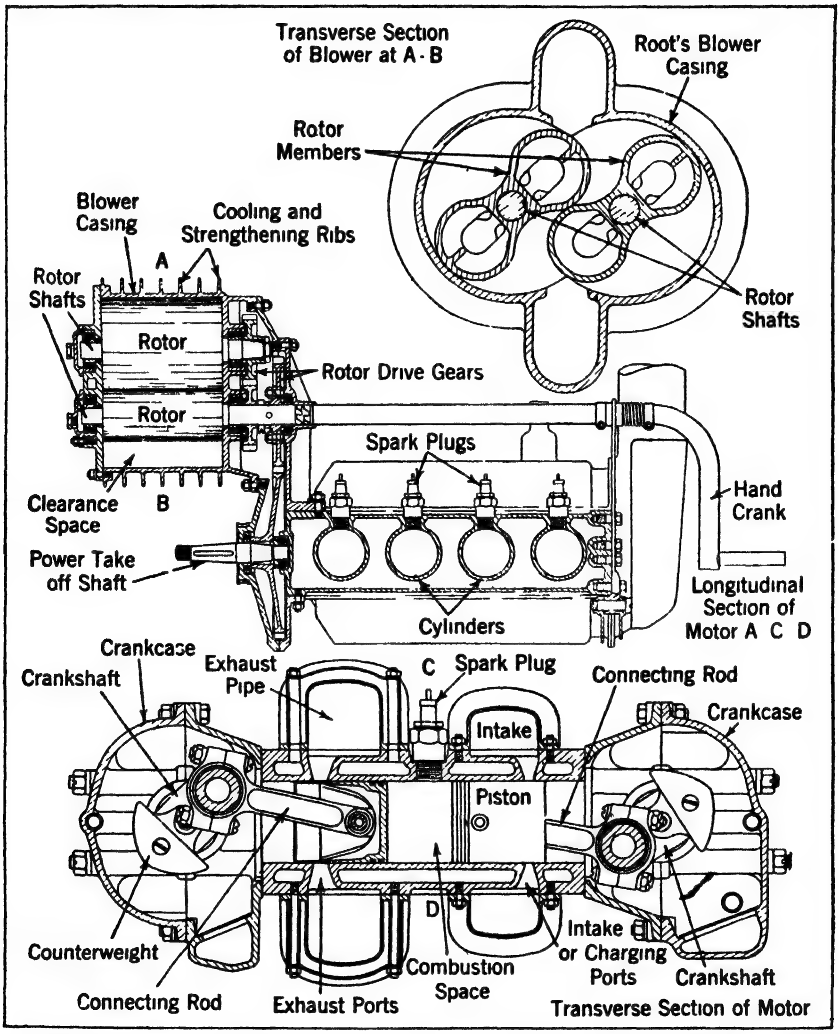


Fig. 25.—Diagrams Showing Construction of the Causan Double Piston Supercharged Two-Cycle Engine.

engines of this principle. The general arrangement of the Causan engine is shown at Fig. 25. Two pistons are used in each cylinder, each driving a crankshaft, the two of which are made to turn in unison by geared connection with a central shaft which delivers the power generated by the engine. One piston controls the exhaust ports, the other piston the ports

through which the fresh gas is forced by the Roots displacement type supercharger driven from the central gear. The sparkplug is mounted in the combustion space, which is formed by the two pistons when they are at the inward end of their strokes. The explosion of the compressed gas pushes the pistons apart, and imparts power to the two crankshafts. The piston controlling the exhaust ports around the cylinder opens the ports before the other opens the intake ports and the exhaust timing is thereby given a lead over the intake opening. Further movement of the pistons, after the exhaust gases start to flow out, results in the intake ports opening and a charge of fresh gas being forced in by the blower rotor. The incoming fresh gas, which has greater than atmospheric pressure forces out practically all the burnt gas and a cleaner and denser charge is present when the pistons start to compress it. The engine shown is a four-cylinder type, giving the same number of impulses per crankshaft revolution as an eight-cylinder four-cycle does. In this engine, which has a bore of 60 mm. and a stroke of 66 mm., 136 horsepower is developed at 4,400 r.p.m. The mechanical friction loss must be higher than the conventional two-cycle and power is also required to drive the blower. The same principle is being experimented with by Junkers for aircraft use, but only air is forced in the combustion space and fuel injection on the Diesel principle permits of the elimination of the electric spark for ignition and the use of fuel oil instead of gasoline with a great reduction in fire risk.

What is Work?—Before we can gain a good conception of the heat engine we must get a clear idea as to the meaning of the word "work" as it is used in science. By "work" is meant mechanical energy; that is, the overcoming of resistance by a force acting through a definite distance. Work is measured by the product of the force-acting and the distance through which it acts. Thus, if a ten pound weight is lifted for one foot a certain amount of work is done. If it is lifted two feet twice the work is done. It can be seen by this definition, that if one merely stands and holds the weight he does no work. Such is, of course, opposed to our feelings in the matter were we holding the weight, but surely if we place the weight on a table and let it lie there we cannot say that the table is working. The unit of work in English speaking countries is the "foot pound," and is equivalent to an amount of work equal to that exerted if one pound is lifted by increments of the foot.

Heat is a Form of Energy.—What is this thing called heat of which we speak so frequently? For ages that question has baffled mankind and it is only within comparatively recent times that the question has been answered with any degree of certainty. According to the modern Kinetic Theory, the molecules, or particles of matter of which everything is composed, are in a state of violent motion or vibration, this motion being somewhat restricted along more or less definite paths in solids and liquids, whereas in gases the molecules move in free paths bounded only by collisions with other molecules. The velocity or violence of this motion is dependent upon the temperature, and increases with an increase in temperature. In general it may be said that the energy which it takes to heat up a given body goes into increasing the kinetic energy of motion of those molecules and into increasing their distance apart. In other words, heat

is a form of energy. Before we go any further it might be well to point out a distinction existing between quantity and intensity of heat. To use a time-worn example there may be more heat in a frozen lake than in the boiling tea-kettle on the stove at home, but the heat is more intense, or in other words at a higher temperature, in the tea-kettle.

Measuring Intensity of Heat.—This leads us to methods of measuring heat. First let us consider the intensity or temperature. For this purpose two scales are in common use, the Fahrenheit and the Centigrade. Just how or why the Fahrenheit scale was adopted no one seems exactly to know, or at least explanations seem to differ. At any rate, on this scale the temperature of melting solidified water has been fixed at 32 degrees, and the temperature of boiling water at sea level at 212 degrees, making 180 degrees between them. Unfortunately this is the scale which is in most common use in this country and in England among the engineering profession, and has led to no end of confusion because all scientific work throughout the world as well as most engineering work in the other countries is conducted in terms of Centigrade scale. On this latter scale the temperature of melting ice is fixed at zero degree and that of boiling water at sea level at 100 degrees. In addition to these two scales there is another scale called the Absolute Scale, which may be measured in terms of either Fahrenheit degrees or Centigrade degrees, and of which more will be said in a later paragraph.

Measuring Amount of Heat.—As opposed to this measurement of the intensity of heat, we have another measure of the quantity or amount of heat. If a certain amount of water is heated so many degrees with a given gas flame in five minutes, it will take two such gas flames to heat twice the amount of water the same number of degrees of temperature in the same time. Thus we say that the amount of heat in the second case is twice that in the first case, although the temperature may have been the same in both cases. It is necessary to have some unit of measure to convey to another person just what is meant by so much heat, just as one needs a measure of distance to tell how far it is from New York to Washington. In English-speaking countries this unit has been called the British Thermal Unit, and is equal to the amount of heat which will just heat one pound of water to one degree Fahrenheit. This unit has been conveniently abbreviated the B.t.u. The Metric unit is the Calory, which is the amount of heat which will heat just one gram of water one degree Centigrade. The reasons for choosing water as the standard were numerous, but the most important reasons were that pure water was easy to obtain, that it was the same the world over, and it had with the exception of a few of the lightest gases, a higher heat absorption capacity per pound than any other substance.

Meaning of Specific Heat.—The Specific Heat is the amount of heat, measured let us say in B.t.u., which one pound of a substance will absorb and just have its temperature raised one degree. The specific heat of water is, therefore, 1.00, while that of most other substances is a fraction inasmuch as their heat holding capacity per pound is less than that of water. If then we have a definite substance, iron for example, which has a specific heat of 0.112, and if we want to raise five pounds of it from the

Thermometer Conversion Table

FAHRENHEIT TO CENTIGRADE

F	C	F	C	F	C	F	C
-40	-40.00	30	-1.11	80	26.67	700	371.11
-38	-38.89	31	-0.56	81	27.22	800	426.67
-36	-37.78	32	0.00	82	27.78	900	482.22
-34	-36.67	33	0.56	83	28.33	1000	537.78
-32	-35.56	34	1.11	84	28.89	1100	593.33
-30	-34.44	35	1.67	85	29.44	1200	648.89
-28	-33.33	36	2.22	86	30.00	1300	704.44
-26	-32.22	37	2.78	87	30.56	1400	760.00
-24	-31.11	38	3.33	88	31.11	1500	815.56
-22	-30.00	39	3.89	89	31.67	1600	871.11
-20	-28.89	40	4.44	90	32.22	1700	926.67
-18	-27.78	41	5.00	91	32.78	1800	982.22
-16	-26.67	42	5.56	92	33.33	1900	1037.78
-14	-25.56	43	6.11	93	33.89	2000	1093.33
-12	-24.44	44	6.67	94	39.44	2100	1148.89
-10	-23.33	45	7.22	95	35.00	2200	1204.44
- 8	-22.22	46	7.78	96	35.56	2300	1260.00
- 6	-21.11	47	8.33	97	36.11	2400	1315.56
- 4	-20.00	48	8.89	98	36.67	2500	1371.11
- 2	-18.89	49	9.44	99	37.22	2600	1426.67
0	-17.78	50	10.00	100	37.78	2700	1482.22
1	-17.22	51	10.56	105	40.55	2800	1537 78
2	-16.67	52	11.11	110	43.33	2900	1593.33
3	-16.11	53	11.67	115	46.11	....	....
4	-15.56	54	12.22	120	48.89	....	....
5	-15.00	55	12.78	125	51.67	....	....
6	-14.44	56	13.33	130	54.44	....	....
7	-13.89	57	13.89	135	57.22	....	....
8	-13.33	58	14.44	140	60.00	....	....
9	-12.78	59	15.00	145	62.78	....	....
10	-12.22	60	15.56	150	65.56	....	....
11	-11.67	61	16.11	155	68.33	....	....
12	-11.11	62	16.67	160	71.11	....	....
13	-10.56	63	17.22	165	73.89	....	....
14	-10.00	64	17 78	170	76.67	....	....
15	- 9.44	65	18.33	175	79.44	....	....
16	- 8.89	66	18.89	180	82.22	....	....
17	- 8.33	67	19.44	185	85.00	....	....
18	- 7.78	68	20.00	190	87.78	....	....
19	- 7.22	69	20 56	195	90.55	....	....
20	- 6.67	70	21.11	200	93.33	....	....
21	- 6.11	71	21.67	212	100.00	....	....
22	- 5.56	72	22.22	225	107.22	....	....
23	- 5.00	73	22.78	250	121.11	....	....
24	- 4.44	74	23.33	275	135.00	....	....
25	- 3.89	75	23.89	300	148.89	....	....
26	- 3.33	76	24.44	350	176.67	....	....
27	- 2.78	77	25.00	400	204.44	....	....
28	- 2.22	78	25.56	500	260.00	....	....
29	- 1.67	79	26.11	600	315.56	....	....

CENTIGRADE TO FAHRENHEIT

C	F	C	F	C	F	C	F
-40	-40.0	5	41.0	40	104.0	1300	2372
-38	-36.4	6	42.8	41	105.8	1400	2552
-36	-32.8	7	44.6	42	107.6	1500	2732
-34	-29.2	8	46.4	43	109.4	1600	2912
-32	-25.6	9	48.2	44	111.2	1700	3092
-30	-22.0	10	50.0	45	113.0	1800	3272
-28	-18.4	11	51.8	46	114.8	1900	3452
-26	-14.8	12	53.6	47	116.6	2000	3632
-24	-11.2	13	55.4	48	118.4	2100	3812
-22	- 7.6	14	57.2	49	120.2	2200	3992
-20	- 4.0	15	59.0	50	122.0	2300	4172
-19	- 2.2	16	60.8	55	131.0	2400	4352
-18	- 0.4	17	62.6	60	140.0	....	....
-17	1.4	18	64.4	65	149.0	....	....
-16	3.2	19	66.2	70	158.0	....	....
-15	5.0	20	68.0	75	167.0	....	....
-14	6.8	21	69.8	80	176.0	....	....
-13	8.6	22	71.6	85	185.0	....	....
-12	10.4	23	73.4	90	194.0	....	....
-11	12.2	24	75.2	95	203.0	....	....
-10	14.0	25	77.0	100	212.0	....	....
- 9	15.8	26	78.8	125	257.0	....	....
- 8	17.6	27	80.6	150	302.0	....	....
- 7	19.4	28	82.4	175	347	....	....
- 6	21.2	29	84.2	200	392	....	....
- 5	23.0	30	86.0	250	482	....	....
- 4	24 8	31	87.8	300	572	....	....
- 3	26.6	32	89.6	400	752	....	....
- 2	28.4	33	91.4	500	932	....	....
- 1	30.2	34	93.2	600	1112	....	....
0	32.0	35	95.0	700	1292	....	....
1	33.8	36	96.8	800	1472	....	....
				900	1652	....	....
2	35.6	37	98.6	1000	1832	....	....
3	37.4	38	100.4	1100	2012	....	....
4	39.2	39	102.2	1200	2192	....	....

In order to convert degrees Fahrenheit into degrees Centigrade, subtract 32, multiply the remainder by 5 and divide by 9. To turn Centigrade into Fahrenheit, multiply the number of degrees Centigrade by 9, divide by 5 and add 32.

Example: To convert 77° Fahrenheit into Centigrade $77-32=45$. $45\times\frac{5}{9}=25°$ C.

To convert $-10°$ Centigrade into Fahrenheit $-10\times\frac{9}{5}=18$ $-18+32=14°$ F.

temperature of the room or about 70° F. to 200° F., a difference of 130°, it is easy to see that we will have to apply 0.112 times five times 130 B.t.u.s. In other words, if we put it in mathematical terms, and let

C equal the specific heat of any given substance
t_1 the initial temperature or temperature before heating
t_2 the final temperature or temperature after heating
Q the quantity of heat put in B.t.u.s
W the weight of the substance heated

it is easy to see that

$$Q = W\ C\ (t_1 - t_2) \tag{1}$$

This simply means in plain everyday English that if we want to find out how much heat it is necessary to put into a substance to raise it a given number of degrees in temperature we merely multiply the weight of the substance by the specific heat and then by the difference of temperature, or the degrees rise.

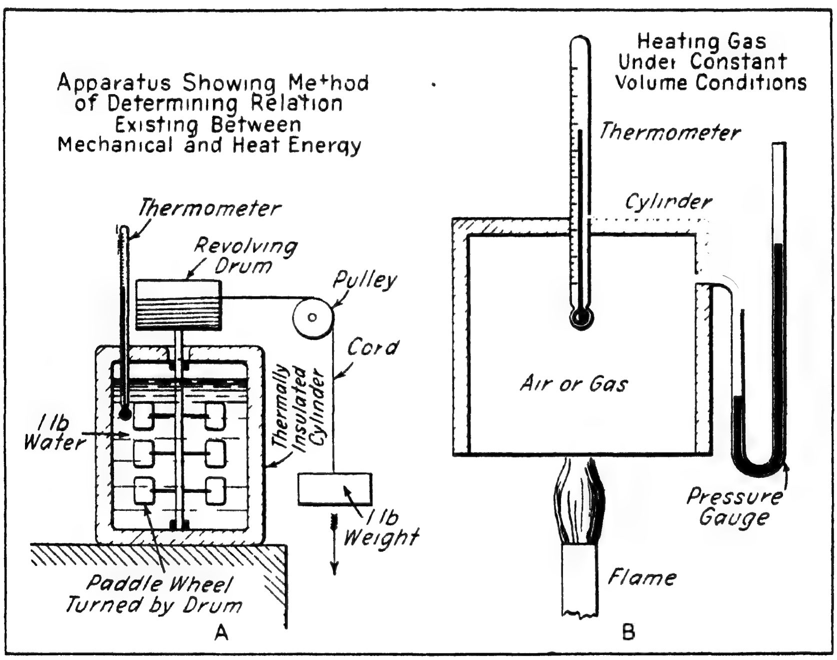


Fig. 26.—Apparatus Showing Method of Determining Relation Existing Between Mechanical and Heat Energy at A. Diagram Showing Heating A Gas Under Constant Volume Conditions at B.

First Law of Thermodynamics.—We all know that if one rubs one's hands together very vigorously they will get warm, in fact very warm, just try it once. How many readers have not, when they were young climbed a rope on a dare, only to be tired out at the top and inclined to slide down rather rapidly, and as a result suffered burned hands and legs where the rope was allowed to slip? How often have we seen the sparks fly from the brake shoes of a long freight train as it was sliding down grade? These are all examples of the working out of the first law of thermodynamics;

that is, that work can be converted into heat. On the other hand, suppose that we put a fire-cracker under a tin can and light the fuse. The can will be blown a hundred feet in the air if we have a large enough fire-cracker. This is an example of the first law of thermodynamics working in the other direction; that is, that heat can be turned into work. The heat generated by the burning and explosion of the powder in the fire-cracker was turned into work in lifting the can up into the air.

Relation of Heat and Work.—Scientifically this proposition is demonstrated in a bit different manner, merely so that quantitative measurements can be made. Thus if an apparatus as in Fig. 26 A is set up, arranged so that as the one pound weight falls it will unwind the cord from the drum and turn the paddles in the water contained in the insulated container, heat will be generated due to the friction of the wheels churning in the water. Now if the can were insulated to prevent the radiation of heat, the pulley nearly frictionless, and if there were just one pound of water in the container, it has been been found that when the weight has dropped about 800 feet the temperature of the water will have increased just about one degree Fahrenheit. Of course, the apparatus as described and illustrated is rather crude but it is presented to show principles. However, the results of very refined measurements indicate that work can be converted into heat in the ratio of 778 foot pounds of work to one B.t.u. of heat.

Conversely it can be shown theoretically that if we had a perfect heat engine, and if we added one B.t.u. to it we should get 778 foot pounds of work out. Unfortunately, we never have actually accomplished this latter feat. The most efficient heat engine known which cannot be applied to aircraft on account of its weight, (The Still Engine) gets but little more than 350 foot pounds out of each B.t.u. put in, whereas most engines get but from 50 to 200 foot pounds of work per B.t.u. added. A formal statement of this first law of thermodynamics is as follows:—Heat can be converted into work, or work into heat in the ratio of 778 foot pounds to one B.t.u. This is merely another way of stating the law of conservation of energy; that is, Energy cannot be created or destroyed. It can, however, be changed from one form to another.

Laws of Gases.—In studying gases it is general practice to speak of so-called "perfect gases." Such a gas does not exist, but all gases that are considerably above their liquefaction temperature behave so very nearly like what we think a perfect gas would do that we call them practically perfect gases. Steam, unless highly superheated, does not, therefore, behave as a perfect gas because it is a water vapor and not a gas.

If we have a closed vessel as in Fig. 26 B, filled with gas and having a thermometer so arranged that the temperature of the gas can be measured, and also a manometer or pressure gauge so arranged that the pressure can be measured, we will find if we heat the gas up, its pressure will rise 1/460 of what it was at 0° F., for every degree that we raise its temperature. This is, of course, only true if we speak of the absolute pressure; that is, the pressure above the atmospheric pressure, which our gauge will record, plus the atmospheric pressure. Thus if we have 5.3 pounds per square inch gauge pressure, or twenty pounds per square inch absolute pressure with the temperature at 0° F., and then if we raise the temperature

to 460° F., it follows that we will double the pressure which we had at zero and will therefore have 40 pounds per square inch absolute or 25. 3 pounds gauge.

Meaning of Absolute Scale.—Now on the other hand, if we have a vessel as shown in Fig. 27 with a frictionless and weightless piston (such is, of course, not physically possible, but other experiments have been made which indicate that the following would be true) and arranged so that we can measure the temperature, we find for every degree Fahrenheit that we raise the temperature we would increase the volume of the gas 1/460 of

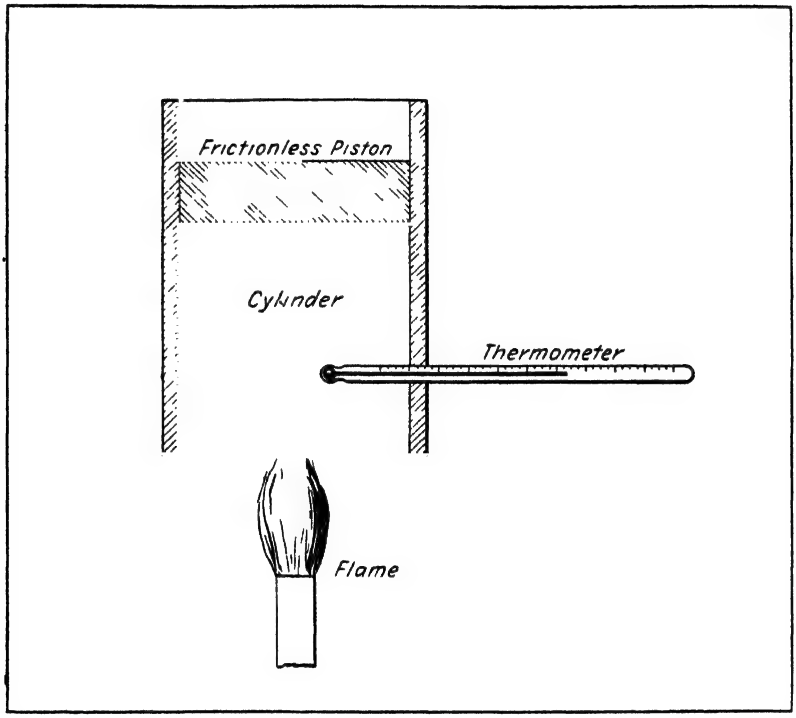


Fig. 27.—Set-Up of Apparatus to Show Expansion of Gas by Increase in Temperature

its volume at 0° F. The piston would, of course move out and the pressure would remain the same; that is, it would always equal the atmospheric pressure on top of the piston. This means that if we have one cubic foot of gas at zero degrees temperature we would have two cubic feet at 460° F. The two facts as given above lead to the conclusion that if we have a perfect gas (one which would obey the above laws at all temperatures) the volume and pressure would become zero when a temperature of 460° F. below zero was attained. This point is called the Absolute Zero. Such could not take place with real gases because the molecules of the gas take up some room, and at very low temperatures all of our gases liquefy or freeze into solids. Hydrogen liquefies at the lowest temperature, and has been boiled at very low pressures within a few degrees of the Absolute Zero. This then gives us a new temperature scale, the so-called Absolute scale which was spoken of previously. The temperature on this scale can

be found by simply adding 460° to any temperature Fahrenheit. Thus ice melts at 460° plus 32° or 492° Absolute on the Fahrenheit scale.

Fundamentals of Thermodynamics.—The above leads us to two very fundamental laws of thermodynamics. Thus it can be said that at constant volume the pressure changes in direct proportion to the absolute temperature. Likewise it can be said that at constant pressure the volume changes in direct proportion to the change in absolute temperature. If T_1 and T_2 represent any initial and final temperatures respectively on the Absolute scale, and P_1 and P_2 represent the corresponding initial and final pressures under constant volume conditions, then

$$\frac{P_1}{P_2} = \frac{T_1}{T_2} \tag{2}$$

or in other terms

$$P = CT \tag{3}$$

where C is a constant depending upon the unit of pressure and temperature. Likewise from the latter law it can be seen that if V_1 and V_2 represent the initial and final volumes corresponding to an initial and final temperature with the pressure remaining constant we will have

$$\frac{V_1}{V_2} = \frac{T_1}{T_2} \tag{4}$$

or in other terms

$$V = CT \tag{5}$$

where C is another constant, different from the one given above, but depending upon the units of V chosen.

We can now derive one of the most important formulas in the thermodynamics of gases. Suppose that we have a cylinder with a piston as in Fig. 27. Let us fix the piston in a given position so that it cannot move. We will then have the air or gas as the case may be under the piston at a definite volume, temperature, and pressure, say V_1, T_1, and P_1. If now we heat the contained gas to some higher temperature, say T_m where T_m is any temperature different from T_1, the pressure will rise let us say to P_2 but the volume will remain the same. If now we release the piston and still heating the gas allow the piston to move out so that the pressure remains just at the pressure P_2, we will finally get a new volume V_2 when the temperature reaches a value which we will call T_2. Now in the first instance we have

$$\frac{P_1}{P_2} = \frac{T_1}{T_m} \text{ or } T_m = \frac{T_1 P_2}{P_1} \tag{6}$$

and in the second instance

$$\frac{V_1}{V_2} = \frac{T_m}{T_2} \text{ or } T_m = \frac{V_1 T_2}{V_2} \tag{7}$$

Things equal to the same thing are equal to each other, so we have

$$\frac{T_1\ P_2}{P_1} = \frac{V_1\ T_2}{V_2} \tag{8}$$

$$\text{or}\ \frac{P_1\ V_1}{T_1} = \frac{P_2\ V_2}{T_2} = R \tag{9}$$

where R is a constant, depending in value upon the gas and the units of volume, pressure, and temperature.

If we are considering one pound of air with P in pounds per square foot and T in degrees absolute on the Fahrenheit scale, then R is equal to 53.36 This simply means that the product of the volume and pressure divided by

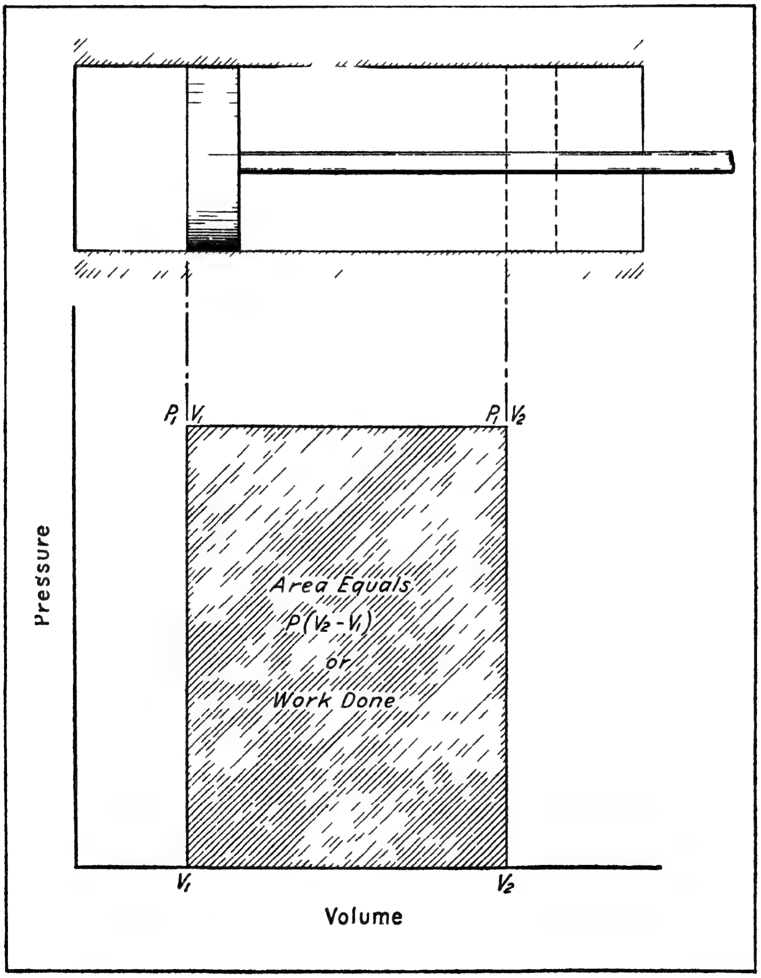


Fig. 28.—Diagram Showing Constant Pressure Expansion of Gas. Distance Along the Vertical Line Indicates Pressure in Pounds per Square Foot, and Length of Horizontal Line Represents Volume in Cubic Feet. The Cross Hatched Area Represents Work Done in Foot-Pounds.

the absolute temperature of any given conditions is equal to the product of its pressure and volume divided by its temperature under any other condition. Thus it can be seen that knowing any two of these conditions enables us to arrive at the third providing we know also the kind of gas since we can then determine the value of R. This same equation is more generally expressed in the form

$$PV = wRT \tag{9a}$$

where w is the weight of the gas in pounds contained in the volume V.

Specific Heat of Gases at Constant Volume.—Gases differ from solids and liquids in that they have two distinctly different specific heats. We found in a previous paragraph that the specific heat is the amount of heat required to raise one pound of a substance 1° F. This heat or energy goes into increasing the speed of motion and thus the kinetic energy of the molecules of which the substance is composed. Suppose now that we have a gas confined in a vessel as shown in Fig. 26 B. If this is heated, it will rise in temperature and pressure, but its volume will remain the same. Under these conditions it will require a definite amount of heat to raise its temperature one degree. For air this amount is about 0.170, depending upon the temperature, and moisture content, which heat, of course, goes into increasing the kinetic energy of the molecules. The specific heat under these conditions is called the "specific heat at constant volume" and is generally designated as C.

Specific Heat at Constant Pressure.—On the other hand, let us consider conditions such as shown in Fig. 28 with the gas confined in a cylinder having a movable piston so that it is held at constant pressure. If now the gas is heated the piston will move out and although the temperature will rise the pressure will remain the same, while the volume will increase. In order to raise the temperature of one pound one degree under these conditions we will, of course, have to supply the same amount of heat as before in order to speed up the movement of the molecules so as to correspond to the new temperature. In addition the piston is forced out of the cylinder against resistance, this means that a definite amount of heat will have to be supplied as before in order to speed up the movement of the molecules so as to correspond to the new temperature. In addition the piston is forced out of the cylinder against resistance, this means that a definite amount of work is done, and since, as we found, heat and work are mutually convertible, the energy necessary to do this work must come from the heat energy supplied to the gas. This means that we must supply an amount of heat in addition to the 0.170 B.t.u. per pound per degree rise which will be equal to the work done in forcing the piston out. It comes out in the case of air at ordinary temperatures that this amount is about 0.070 B.t.u. making the total amount of heat which must be supplied to raise one pound of air one degree Fahrenheit under constant pressure conditions about 0.240 B.t.u. This is called the "specific heat at constant pressure" and is generally designated as C_p.

Constant Pressure Expansion.—Suppose we have a cylinder as shown at Fig. 28, with a frictionless piston as before and let us assume, also, that we

have a rod attached to the piston and moving with it so as to transmit the pressure and thus the work to some kind of a machine. Let us assume, further, that we have a gas confined in the cylinder at the condition of pressure and volume as represented on the curve by P_1V_1. (The horizontal scale represents volumes, the vertical scale represents pressure.) If now we apply heat as before and keep the pressure on the piston constant, the piston will move out as the temperature rises to some other volume V_2 at pressure P_1. This expansion is indicated upon the pressure-volume diagram by the line P_1V_1 to P_1V_2. Now the work done on the piston is equal to the pressure on the piston times the distance in feet moved, but the total pressure is equal to the pressure per square foot of area times the total area of the piston in square feet. In other words, then, if we let d equal the distance through which the piston moves, P the pressure in pounds per square foot, and A the area of the piston in square feet, for the work done or W we will have

$$W=PAd \tag{10}$$

but Ad is equal to the increase in volume or to $(V_2—V_1)$. Therefore

$$W=P\ (V_2—V_1). \tag{11}$$

Now if the ordinate or vertical scale on the curve represents pressures in pounds per square foot, and the horizontal scale represents cubic feet volume of the gas in the cylinder, then the cross-hatched area represents $P\ (V_2—V_1)$, or in other words the amount of work done in foot-pounds. It can also be shown by means of the calculus that the area under any expansion line on the P-V diagram, be it curved or otherwise, represents the work done. It is this principle which is the fundamental basis of the steam and gas engine indicator.

Constant Temperature or Isothermal Expansion.—It has been found experimentally that if the temperature of a gas be kept constant and its volume doubled its pressure will be just one-half what it was before the change in volume took place. Likewise if its volume be increased four times, its pressure becomes one-fourth, and so on. In other words the product of the pressure and the volume of a gas at constant temperature is always constant. That is PV=C. This can be seen from equation (9a), for we found there that PV=wRT, where wRT becomes the constant C. Such expansion is called isothermal, meaning constant temperature, and must be represented on the P-V diagram by a rectangular hyperbola as shown in Fig. 29. From the preceding paragraph we saw that the area under any curve is equal to the work done during the expansion. Since the gas is at the same temperature and thus has the same energy in it at the end of the expansion as it had at the beginning, we reach the inevitable conclusion that heat must have been put into the gas during the expansion in order to account for the work done. By actual experiment we find this to be true, and the heat added is just equivalent to the work done. Thus we have very nearly a perfect heat engine for a part of one cycle, but the trouble comes when we attempt to raise the pressure back again so that we can do it over again.

Adiabatic Expansion.—We have one other fundamental type of expansion called Adiabatic expansion. This means expansion without the

addition or subtraction of heat as heat. Suppose that we have the cylinder and piston as in Fig. 29, made of a perfect heat insulating material. If now we have a gas confined in the cylinder at P_1V_1 conditions and allow it to expand to some condition P_2V_2 we will have an amount of work done equal to the area under the expansion curve. On account of the insulating

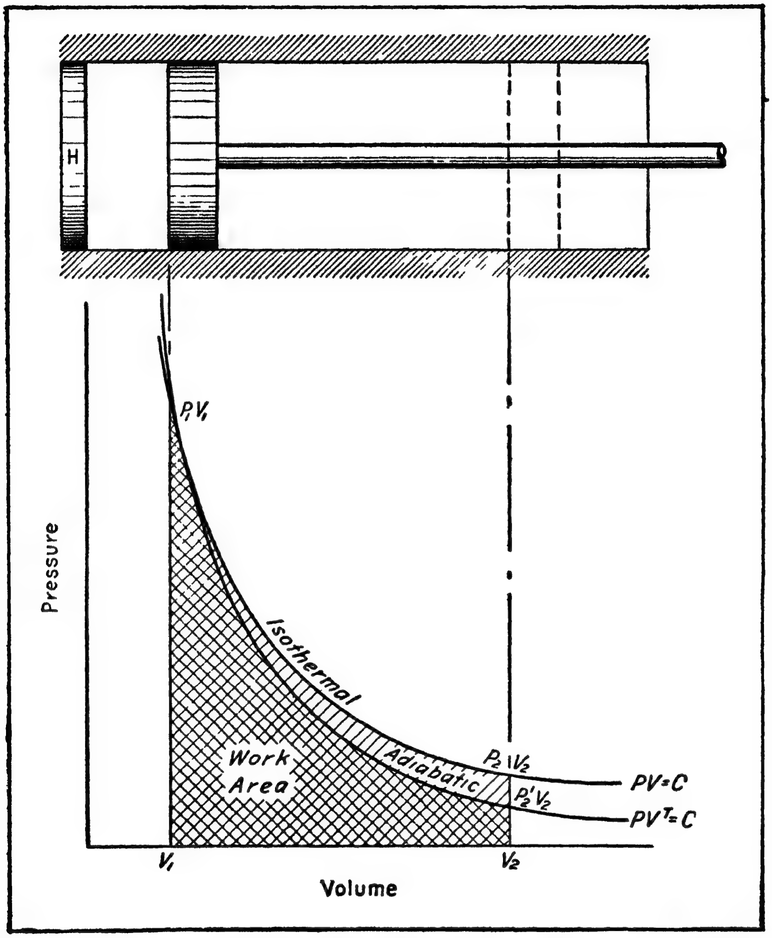


Fig. 29.—Isothermal and Adiabatic Expansion of Gas. The Vertical and Horizontal Scale Represents the Same Quantities as in Fig. 28. The Cylinder Must be a Perfect Conductor in the Case of Isothermal Expansion in Order to Keep the Gas at the Same Temperature as the Surroundings and it Must be a Perfect Insulator in the Case of Adiabatic Expansion, Impossible in Practice but Only Assumed for Theoretical Consideration.

side walls and piston we will not, however, have added any heat from an external source. This means that the energy necessary to do the external work must come from the internal energy or heat of the gas itself, and consequently its temperature must drop. This accounts for the lower pressure at any given volume on the adiabatic curve. The equation of this curve is

$$PV^r = C$$

where C is a constant, and r, the exponent of V, is a constant for any particular gas at a given temperature, and is equal to

$$\frac{C_p}{C_v}$$ or about 1.41 for air at ordinary temperatures.

To derive the foregoing equation requires the use of the calculus and its use requires a knowledge of logarithms or the slide rule. Those interested in following this farther will find complete instructions and the derivation in any standard text book on Thermodynamics.

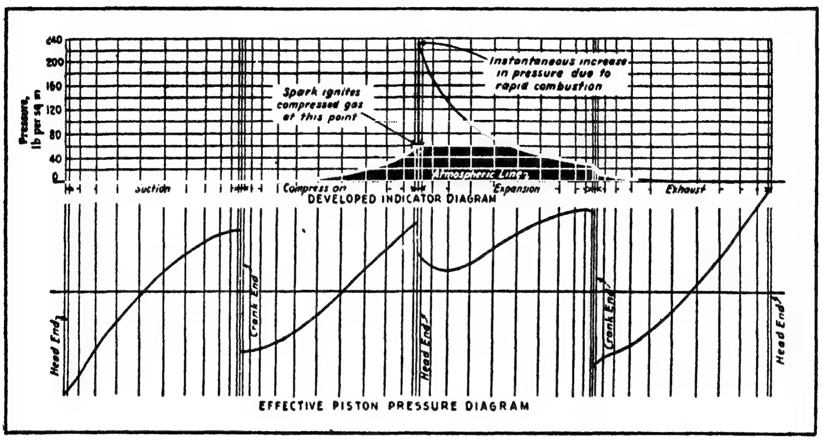


Fig. 30.—Drawings Showing Effective Piston Pressure as Shown in Well Developed Indicator Diagram.

In all of the three fundamental types of expansion which we have just considered, we might have compressed the gas instead of expanding it. We would then have had to expend work and to have subtracted heat instead of adding heat. That is the gas would have become heated during compression due to the work done upon it by the machine furnishing the power for compression. It is this phenomena which makes the bicycle pump get hot when we work it rapidly and which calls for air- or water-cooling cylinders of air pumps that are used continuously.

Actual Expansion Curves.—In actual practice in heat engine or air compressor work we very seldom have either pure constant pressure, isothermal, or adiabatic expansion or compression. The actual expansion curves approach these theoretical curves if the conditions are right, but due to friction of valves, heat flow through walls, and other factors the theoretical curves are very seldom followed exactly. However, a knowledge of these three fundamental curves is necessary to fully understand the various heat engine cycles. The writer can imagine the reader who has studied the foregoing, saying that all of this matter on heat is very interesting but what has it got to do with aviation engines and their basis of operation. Aviation engines are a form of heat engine and it is by the

pressure produced on the pistons due to the expansion of the ignited gas charge that heat energy is converted into mechanical energy. Losses of heat are inevitable in this conversion process and the engine that loses the least heat is the most efficient type.

The effective piston pressure diagram at Fig. 30 shows how an indicator diagram can be developed and the pressure on the piston of a four-stroke cycle gas-engine can be shown graphically. The vertical scale shows pressure in pounds per square inch. It will be seen that during the suction stroke of the piston and for a short period on the compression stroke the pressure is less than atmospheric, rising to about 60 pounds just prior to

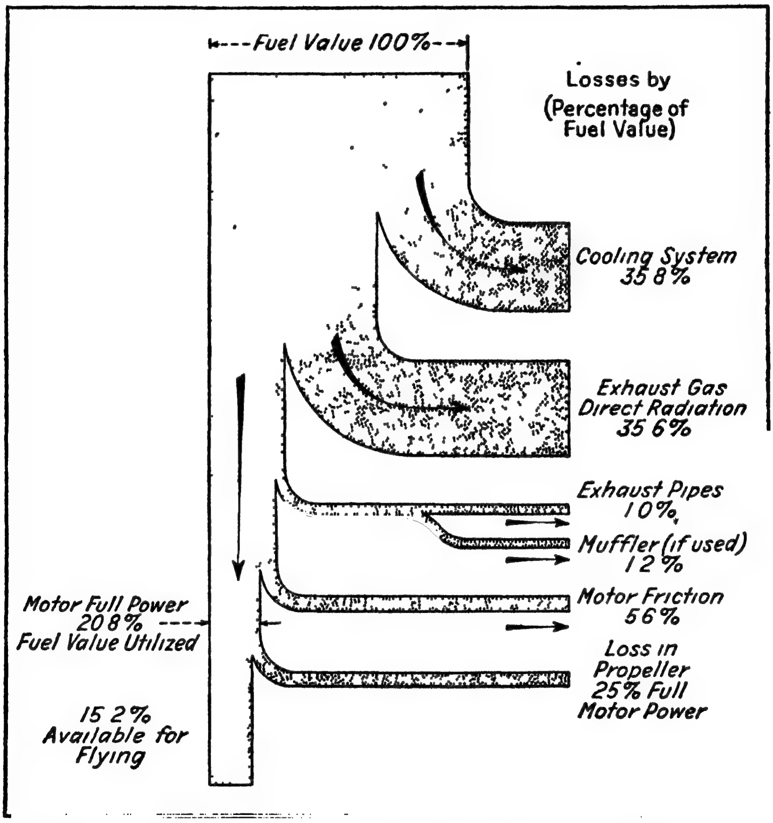


Fig. 31.—Graphic Diagram Showing Amount of Heat Energy Obtained by Combustion of Fuel Actually Utilized for Flying in the Form of Propeller Thrust in Average Water-Cooled Gasoline Engine.

ignition, which takes place at the end of the compression stroke. The rise in pressure is very rapid, reaching nearly 240 pounds per square inch due to the explosion. This is four times the compression pressure indicated. During the expansion stroke the pressure drops to about 30 pounds per square inch at which point the exhaust valve opens and during the following exhaust or scavenging stroke the pressure drops to atmospheric as

indicated by the zero line. A proper understanding of the fundamental heat laws will enable the student to easily follow discussions of heat engine theory and will enable one to make comparisons between different engine types as relate to thermal efficiency as well as understanding the more visible and easily understood points involving the mechanism of the engines and their mechanical efficiency.

Graphic Representation of Fuel Efficiency.—As an example, the fuel efficiency, or actual proportion of the total B.t.u. contained in the fuel burned that is effectively utilized at the propeller for flying is given at Fig. 31. Starting with 100%, we find various inevitable losses, and while the figures given will vary with different types of engines and their installation they are valuable as a guide and the values given may be considered typical for a water-cooled engine. The heat loss through the jackets may vary and in engines exhausting directly into the air, the muffler losses may be eliminated, or at least, greatly reduced, though any form of gas collector ring or exhaust manifold will introduce power losses varying in degree with the back pressure.

QUESTIONS FOR REVIEW

1. What is a four-cycle engine?
2. Name two types of two-cycle engines and indicate how they differ.
3. What causes fresh gas to fill engine cylinders?
4. What is the function of the exhaust stroke?
5. What advantages are obtained by supercharging in two-cycle engines?
6. What is heat; work; energy?
7. What is the relation of heat and work?
8. Outline the first law of gases.
9. What is absolute temperature?
10. What is the difference between isothermal and adiabatic expansion?

CHAPTER III

THEORY OF HEAT ENGINES—DIESEL ENGINES

Theory of the Heat Engine—Early Gas Engine Forms—The Isothermal Law—The Adiabatic Law—Temperature Computations—Heat Energy Converted to Mechanical Work—Results of Impure Charge—Tests With Air and Gas Mixtures—Efficiency of Conversion of Heat to Power—Requisites for Best Power Effect—Thermodynamics of Aircraft Engines—Heat Loss to Cooling System—The Combustion Process—Turbulence of Great Value—Stratified Charge—Detonation Prevention Important—Diesel Engines for Aircraft—Four-Stroke Diesel Engine Action—Two-Stroke Diesel Engine Action—Port Scavenging Type—Overhead Valve Scavenging Type—High Speed Diesel Engines—Marine Diesel Engines Very Heavy—Diesel Mean Effective Pressure Low—Liquid Fuel Atomization a Problem—Aircraft Diesel Engines Not Yet Available—The Attendu Solid Injection Oil Engine—Elements of the Attendu Fuel System—Results of Navy Tests.

The laws controlling the elements that create a power by their expansion by heat due to combustion, when properly understood, become a matter of computation in regard to their value as an agent for generating power in the various kinds of explosive engines. The method of heating the elements of power in explosive engines greatly widens the limits of temperature as available in other types of heat engines. It disposes of many of the practical troubles of hot-air, and even of steam-engines, in the simplicity and directness of application of the elements of power. In the explosive engine the difficulty of conveying heat for producing expansive effect by convection is displaced by the generation of the required heat within the cylinder containing the expansive element and at the instant of its useful work. The low conductivity of heat to and from air has been the great obstacle in the practical development of the hot-air engine, now obsolete as a source of power and only seen as an interesting scientific toy by the present generation, and the waste of heat in steam boiler flues also accounts for the low thermal efficiency of the steam-engine. On the contrary, it has become the source of economy and practicability in the development of the internal-combustion engine.

Theory of the Heat Engine.—The action of air, gas, and the vapors of gasoline and petroleum oil, whether singly or mixed, is affected by changes of temperature practically in nearly the same ratio; but when the elements that produce combustion are interchanged in confined spaces, there is a marked difference of effect. The oxygen of the air, the hydrogen and carbon of a gas, or vapor of gasoline or petroleum oil are the elements that by combustion produce heat to expand the nitrogen of the air and the watery vapor produced by the union of the oxygen in the air and the hydrogen in the gas, as well as also the monoxide and carbonic-acid gas that may be formed by the union of the carbon of gas or vapor with part of the oxygen of the air. The various mixtures as between air and gas, or air and vapor, with the proportion of the products of combustion left in the cylinder from a previous combustion, form the elements to be considered in estimating the amount of pressure that may be obtained by their combustion and expansive force.

Miscellaneous Engine Terms Defined.—The following terms used in this treatise have been defined by the National Advisory Committee for Aeronautics and are used in the sense given by the standard definitions.

brake mean effective pressure—The net unit pressure which, if applied during the power strokes to the pistons of an engine having no mechanical losses, would produce the given brake horsepower at the stated speed.

dry weight of an engine—The weight of the engine, including carburetor and ignition systems complete, propeller hub assembly, reduction gears, if any, but excluding exhaust manifolds, oil and water. If the starter is built into the engine as an integral part of the structure, its weight shall be included.

fixed powerplant weight for a given airplane—The weight of the engine, including ignition, carburetor and induction systems complete, propeller and hub, exhaust manifolds, radiator and water, *if used,* with all interconnecting wires, controls, tanks, and pipes, lubricating oil temperature regulators, *the oil contained in the engine crankcase,* and the starting gear attached to the engine, but excluding fuel, oil, and engine instruments.

maximum horsepower of an engine—The maximum horsepower which an engine can develop.

maximum revolutions—The number of revolutions per minute corresponding to the maximum horsepower.

rated horsepower of an engine—The average horsepower developed by an engine of a given type in passing the standard 50-hour endurance test.

rated revolutions—The number of revolutions corresponding to the rated horsepower.

specific fuel (or oil) consumption—The weight of fuel (or oil) consumed per brake horsepower-hour.

weight per horsepower—The dry weight of an engine divided by the rated horsepower.

Early Gas-Engine Forms.—The working process of the explosive motor may be divided into three principal types: 1. Motors with charges igniting at constant volume without compression, such as the Lenoir, Hugon, and other similar types now abandoned because they were so wasteful in fuel and power effect. 2. Motors with charges igniting at constant pressure with compression, in which a receiver is charged by a pump and the gases burned while being admitted to the motor cylinder, such as types of the Simon and Brayton engine also obsolete. 3. Motors with charges igniting at constant volume with variable compression, such as the later two- and four-cycle motors with compression of the indrawn charge; limited to some extent in the two-cycle type and variable in the four-cycle type with the ratios of the clearance space in the cylinder. This principle produces the explosive motor of greatest efficiency.

The phenomena of the brilliant light and its accompanying heat at the moment of explosion have been witnessed in the experiments of Dugald Clerk in England, the illumination lasting throughout the stroke; but in regard to time in a four-cycle engine, the incandescent state exists only

one-quarter of the running time. Thus the time interval, together with the nonconductibility of the gases, makes the phenomena of a very high-temperature combustion within the comparatively cool walls of a metal cylinder a practical possibility because the metal has time to cool off between explosions even though the time interval be extremely small.

The Isothermal Law.—The natural laws, long since promulgated by Boyle, Gay Lussac, and others, on the subject of the expansion and compression of gases by force and by heat, and their variable pressures and temperatures when confined, are conceded to be practically true and applicable to all gases, whether single, mixed, or combined as the writer has previously pointed out.

The law formulated by Boyle only relates to the compression and expansion of gases without a change of temperature, and is stated in these words:

If the temperature of a gas be kept constant, its pressure or elastic force will vary inversely as the volume it occupies.

It is expressed in the formula $P \times V = C$, or pressure $\times$ volume = constant. Hence, $\frac{C}{P} = V$ and $\frac{C}{V} = P$.

Thus the curve formed by increments of pressure during the expansion or compression of a given volume of gas without change of temperature is designated as the isothermal curve in which the volume multiplied by the pressure is a constant value in expansion, and inversely the pressure divided by the volume is a constant value in compressing a gas.

But as compression and expansion of gases require the expenditure of energy or force for their accomplishment mechanically, or by the application or abstraction of heat chemically or by convection; a second condition becomes involved, which was formulated into a law of thermodynamics by Gay Lussac under the following conditions: A given volume of gas under a free piston expands by heat and contracts by the loss of heat, its volume causing a proportional movement of a free piston equal to $^1/_{273}$ part of the cylinder volume for each degree Centigrade difference in temperature, or $^1/_{492}$ part of its volume for each degree Fahrenheit. With a fixed piston (constant volume), the pressure is increased or decreased by an increase or decrease of heat in the same proportion of $^1/_{273}$ part of its pressure for each degree Centigrade, or $^1/_{492}$ part of its pressure for each degree Fahrenheit change in temperature. This is the natural sequence of the law of mechanical equivalent, which is a necessary deduction from the principle that nothing in nature can be lost or wasted, for all the heat that is imparted to or abstracted from a gaseous body must be accounted for, either as heat or its equivalent transformed into some other form of energy. In the case of a piston moving in a cylinder by the expansive force of heat in a gaseous body, all the heat expended in expansion of the gas is turned into work; the balance must be accounted for in absorption by the cylinder or radiation, as clearly shown at Fig. 31. The small amount that is converted into mechanical energy at the propeller is apparent.

The Adiabatic Law.—This theory is equally applicable to the cooling of gases by abstraction of heat or by cooling due to expansion by the motion of a piston. The denominators of these heat fractions of expansion or contraction represent the absolute zero of cold below the freezing-point of water, and read —273° C. or —492.66° = —460.66° F. below zero; and, as previously explained, these are the starting-points of reference in computing the heat expansion in gas-engines. According to Boyle's law, called the first law of gases, there are but two characteristics of a gas and their variations to be considered, *viz.*, volume and pressure: while by the law of Gay Lussac, called the second law of gases, a third is added, consisting of the value of the absolute temperature, counting from absolute zero to the temperatures at which the operations take place. This is the *Adiabatic* law.

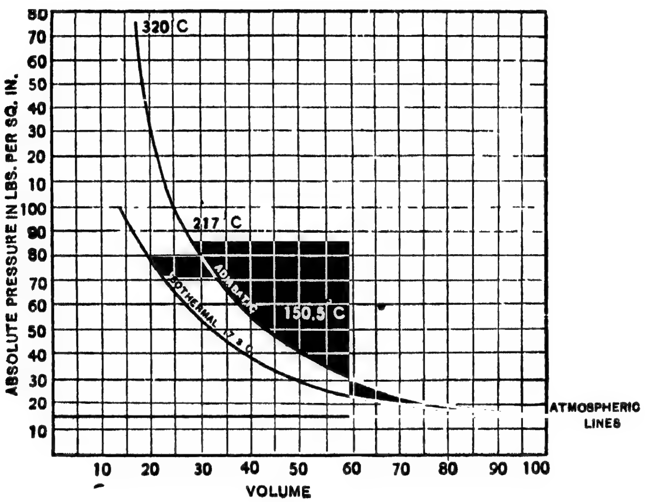


Fig. 32.—Diagram Isothermal and Adiabatic Lines.

The ratio of the variation of the three conditions—volume, pressure, and heat—from the absolute zero temperature has a certain rate, in which the volume multiplied by the pressure and the product divided by the absolute temperature equals the ratio of expansion for each degree. If a volume of air is contained in a cylinder having a piston and fitted with an indicator, the piston, if moved to and fro slowly, will alternately compress and expand the air, and the indicator pencil will trace a line or lines upon the card, which lines register the change of pressure and volume occurring in the cylinder. If the piston is perfectly free from leakage, and it be supposed that the temperature of the air is kept quite constant, then the line so traced is called an *Isothermal line*, and the pressure at any point when multiplied by the volume is a constant, according to Boyle's law,

$$pv = \text{a constant.}$$

If, however, the piston is moved very rapidly, the air will not remain at constant temperature, but the temperature will increase because work has been done upon the air, and the heat has no time to escape by conduction.

If no heat whatever is lost by any cause, the line will be traced over and over again by the indicator pencil, the cooling by expansion doing work precisely equalling the heating by compression. This is the line of no transmission of heat, therefore known as *Adiabatic.*

The expansion of a gas $^1/_{273}$ of its volume for every degree Centigrade, added to its temperature, is equal to the decimal .00366, the coefficient of expansion for Centigrade units. To any given volume of a gas, its expansion may be computed by multiplying the coefficient by the number of degrees, and by reversing the process the degree of acquired heat may be obtained approximately. These methods are not strictly in conformity with the absolute mathematical formula, because there is a small increase in the increment of expansion of a dry gas, and there is also a slight difference in the increment of expansion due to moisture in the atmosphere and to the vapor of water formed by the union of the hydrogen and oxygen in the combustion chamber of explosive engines.

Temperature Computations.—The ratio of expansion on the Fahrenheit scale is derived from the absolute temperature below the freezing-point of water (32°) to correspond with the Centigrade scale; therefore $\frac{1}{492.66}$ — .0020297, the ratio of expansion from 32° for each degree rise in temperature on the Fahrenheit scale. As an example, if the temperature of any volume of air or gas at constant volume is raised, say from 60° to 2,000° F., the increase in temperature will be 1,940°. The ratio will be $\frac{1}{520.66} = .0019206$. Then by the formula:

Ratio × acquired temp. × initial pressure = the gauge pressure; and .0019206 × 1,940° × 14.7 = 54.77 lbs.

By another formula, a convenient ratio is obtained by $\frac{\text{absolute pressure}}{\text{absolute temp.}}$ or $\frac{14.7}{520.66} = .028233$; then, using the difference of temperature as before, .028233 × 1,940° = 54.77 lbs. pressure.

By another formula, leaving out a small increment due to specific heat at high temperatures:

I. $\frac{\text{Atmospheric pressure} \times \text{absolute temp.} + \text{acquired temp.}}{\text{Absolute temp.} + \text{initial temp.}}$ = absolute pressure due to the acquired temperature, from which the atmospheric pressure is deducted for the gauge pressure. Using the foregoing example, we have $\frac{14.7 \times 460.66° + 2{,}000°}{460.66 + 60°} = 69.47 - 14.7 = 54.77$. the gauge pressure,

460.66 being the absolute temperature for zero Fahrenheit.

For obtaining the volume of expansion of a gas from a given increment of heat, we have the approximate formula:

$$\text{II.}\ \frac{\text{Volume} \times \text{absolute temp.} + \text{acquired temp.}}{\text{Absolute temp.} + \text{initial temp.}} = \text{heated volume.}$$

In applying this formula to the foregoing example, the figures become:

$$\text{I.} \times \frac{460.66^\circ + 2{,}000^\circ}{460.66 + 60^\circ} = 4.72604 \text{ volumes.}$$

From this last term the gauge pressure may be obtained as follows:

III. $4.72604 \times 14.7 = 69.47$ lbs. absolute — 14.7 lbs. atmospheric pressure $= 54.77$ lbs. gauge pressure; which is the theoretical pressure due to heating air in a confined space, or at constant volume from 60° to 2,000° F.

By inversion of the heat formula for absolute pressure we have the formula for the acquired heat, derived from combustion at constant volume from atmospheric pressure to gauge pressure plus atmospheric pressure as derived from Example I., by which the expression

$$\frac{\text{absolute pressure} \times \text{absolute temp.} + \text{initial temp.}}{\text{initial absolute pressure}}$$

= absolute temperature + temperature of combustion, from which the acquired temperature is obtained by subtracting the absolute temperature.

Then, for example, $\dfrac{69.47 \times 460.66 + 60}{14.7} = 2{,}460.66$, and $2{,}460.66 - 460.66 = 2{,}000^\circ$, the theoretical heat of combustion. The dropping of terminal decimals makes a small decimal difference in the result in the different formulas.

Heat Energy Converted to Mechanical Work.—By Joule's law of the mechanical equivalent of heat, whenever heat is imparted to an elastic body, as air or gas, energy is generated and mechanical work produced by the expansion of the air or gas. When the heat is imparted by combustion within a cylinder containing a movable piston, the mechanical work becomes an amount measurable by the observed pressure and movement of the piston. The heat generated by the explosive elements and the expansion of the noncombining elements of nitrogen and water vapor that may have been injected into the cylinder as moisture in the air, and the water vapor formed by the union of the oxygen of the air with the hydrogen of the gas, all add to the energy of the work from their expansion by the heat of internal combustion. As against this, the absorption of heat by the walls of the cylinder, the piston, and cylinder-head or clearance walls, becomes a modifying condition in the force imparted to the moving piston.

It is found that when any explosive mixture of air and gas or hydrocarbon vapor is fired, the pressure falls far short of the pressure computed

from the theoretical effect of the heat produced, and from gauging the expansion of the contents of a cylinder. It is now well known that in practice the high efficiency which is promised by theoretical calculation is never realized; but it must always be remembered that the heat of combustion is the real agent, and that the gases and vapors are but the medium for the conversion of inert elements of power into the activity of energy by their chemical union. The theory of combustion has been the leading stimulus

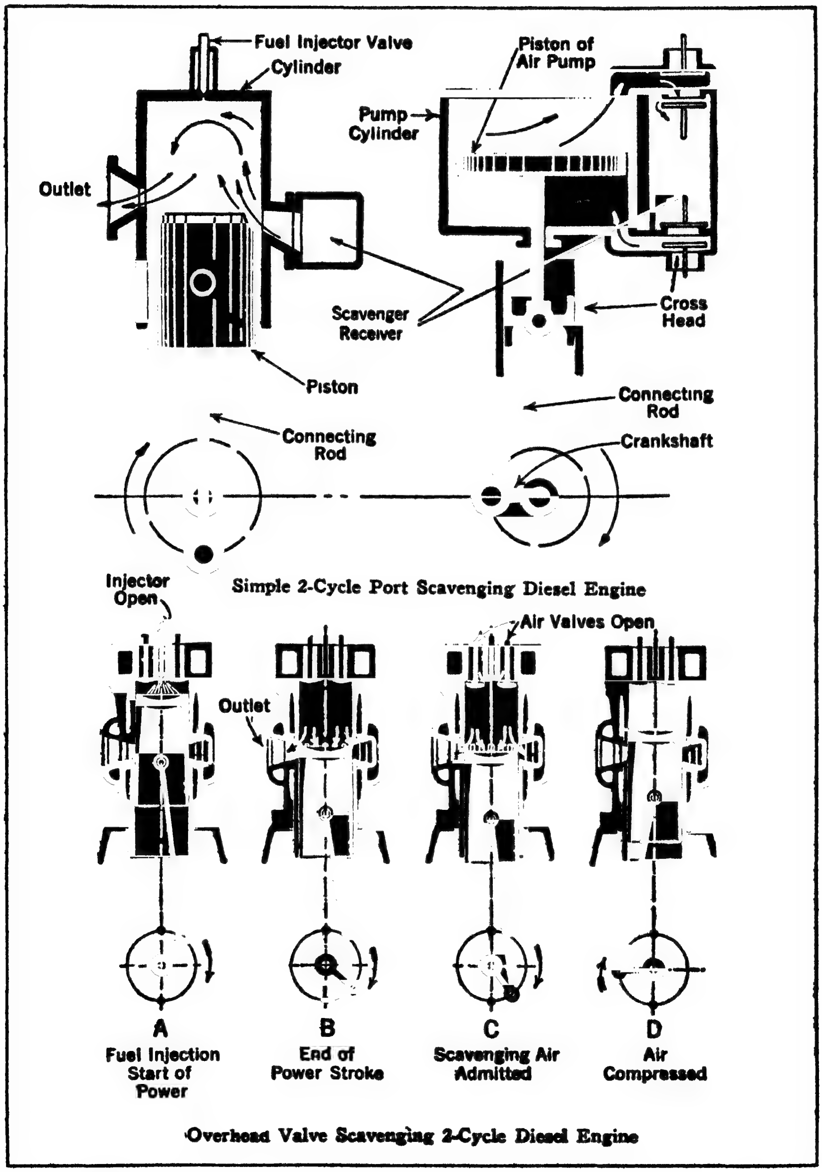


Fig. 32A.—Diagrams Explaining and Comparing Action of Two-Cycle Diesel Engines with Port and Valve Scavenging.

to large expectations with inventors and constructors of explosive motors; its entanglement with the modifying elements in practice has often delayed the best development in construction, and as yet no really positive design of best form or action seems to have been accomplished, although great progress has been made during the past decade in the development of speed, reliability, economy, and power output of the individual units of this now universally applicable prime mover.

Results of Impure Charge.—One of the most serious difficulties in the practical development of pressure, due to the theoretical computations of the pressure value of the full heat, is probably caused by imparting the heat of the fresh charge to the balance of the previous charge that has been cooled by expansion from the maximum pressure to near the atmospheric pressure of the exhaust. The retardation in the velocity of combustion of perfectly mixed elements is now well known from experimental trials with

TABLE I

Explosion at Constant Volume in a Closed Chamber

Mixture Injected	Temp. of Injection Fahr.	Time of Explosion. Second	Observed Gauge Pressure. Pounds	Computed Temp. Fahr.
1 volume gas to 14 volumes air	64°	0.45	40.	1,483°
1 " " " 13 " "	51°	0.31	51.5	1,859°
1 " " " 12 " "	51°	0.24	60.	2,195°
1 " " " 11 " "	51°	0.17	61.	2,228°
1 " " " 9 " "	62°	0.08	78.	2,835°
1 " " " 7 " "	62°	0.06	87.	3,151°
1 " " " 6 " "	51°	0.04	90.	3,257°
1 " " " 5 " "	51°	0.055	91.	3,293°
1 " " " 4 " "	66°	0.16	80.	2,871°

measured quantities; but the principal difficulty in applying these conditions to the practical work of an explosive engine where a necessity for a large clearance space cannot be obviated, is in the inability to obtain a maximum effect from the imperfect mixture and the mingling of the products of the last explosion with the new mixture, which produces a clouded condition that makes the ignition of the mass irregular or chattering, as observed in the expansion lines of indicator cards; but this must not be confounded with the reaction of the spring in the indicator or irregularities due to the action of other parts of the indicator mechanism.

Stratification of the mixture has been claimed as taking place in the clearance chamber of the cylinder by some authorities; but this is not considered a satisfactory explanation by others in view of the vortical effect of the violent injection of the air and gas or vapor mixture. It certainly cannot become a perfect mixture in the time of a stroke of a high-speed motor of the two-cycle class. In a four-cycle engine, making 1,500 revolutions per minute, the injection and compression in any one cylinder take place in one twenty-fifth of a second—formerly considered far too short a time

for a perfect infusion of the elements of combustion but now very easily taken care of despite the extremely high speed of numerous aviation and automobile powerplants. When we consider that high-speed automobile racing engines have exceeded 8,000 r.p.m. it will be seen that the injection and compression in any one cylinder must take place in one-hundredth of a second or even less time. The preceding table shows the results of some early experiments and is of interest because it gives comparative values for different mixtures and compression pressures.

Tests with Air and Gas Mixtures.—In an examination of the times of explosion and the corresponding pressures in both tables Nos. 1 and 2 it will be seen that a mixture of one part gas to six parts air is the most effective and will give the highest mean pressure in a gas-engine. There is a limit to the relative proportions of illuminating gas and air mixture that is explosive, somewhat variable, depending upon the proportion of hydrogen in the gas. With ordinary coal-gas, one of gas to fifteen parts of air; and on the lower end of the scale, one volume of gas to two parts air, are nonexplosive. With gasoline vapor the explosive effect ceases at one to sixteen, and a saturated mixture of equal volumes of vapor and air will not explode, while the most intense explosive effect is from a mixture of one part vapor to nine parts air. In the use of gasoline and air mixtures from a carburetor, the best effect is from one part saturated air to eight parts free air. These figures were obtained by tests made under certain specific conditions and values given may vary in tests made under other conditions and with a different set up of apparatus.

TABLE II

Properties and Explosive Temperature of a Mixture of One Part of Illuminating Gas of 660 Thermal Units per Cubic Foot with Various Proportions of Air Without Mixture of Charge with the Products of a Previous Explosion

Proportion, Air to Gas, by Volumes	Pounds in One Cubic Foot of Mixture	Specific Heat. Heat Units Required to Raise 1 Lb. 1 Deg. Fahrenheit		Heat to Raise One Cubic Foot of Mixture 1 Deg. Fahr.	Heat Units Evolved by Combustion	Ratio Col. $\frac{6}{5}$	Usual Combustion Efficiency	Usual Rise of Temperature due to Explosion at Constant Volume
		Constant Pressure	Constant Volume					
6 to 1...	.074195	.2668	.1913	.014189	94.28	6644.6	.465	3090
7 to 1...	.075012	.2628	.1882	.014116	82.	5844.4	.518	3027
8 to 1....	.075647	.2598	.1858	.014059	73.33	5216.1	.543	2832
9 to 1...	.076155	.2575	.1846	.014013	66.	4709.9	.56	2637
10 to 1...	.076571	.2555	.1825	.013976	60.	4293.	.575	2468
11 to 1...	.076917	.2540	.1813	.013945	55.	3944.	.585	2307
12 to 1...	.077211	.2526	.1803	.013922	50.77	3646.7	.58	2115

The weight of a cubic foot of gas and air mixture as given in Col. 2 Table 2 is found by adding the number of volumes of air multiplied by its weight, .0807, to one volume of gas of weight .035 pound per cubic foot and dividing by the total number of volumes; for example, as in the table, $6 \times .0807 = \frac{.5192}{7} = .074195$ as in the first line, and so on for any mixture or for other gases of different specific weight per cubic foot. The heat units evolved by combustion of the mixture (Col. 6) are obtained by dividing the total heat units in a cubic foot of gas by the total proportion of the mixture, $\frac{660}{7} = 94.28$ as in the first line of the table. Col. 5 is obtained by multiplying the weight of a cubic foot of the mixture in Col. 2 by the specific heat at a constant volume (Col. 4), $\frac{\text{Col. 6}}{\text{Col. 5}} = \text{Col. 7}$ the total heat ratio, of which Col. 8 gives the usual combustion efficiency — Col. 7 × Col. 8 gives the absolute rise in temperature of a pure mixture, as given in Col. 9.

The many recorded experiments made to solve the discrepancy between the theoretical and the actual heat development and resulting pressures in the cylinder of an explosive motor, to which much discussion has been given as to the possibilities of dissociation and the increased specific heat of the elements of combustion and noncombustion, as well, also, of absorption and radiation of heat, have as yet furnished no absolutely satisfactory conclusion as to what really takes place within the cylinder walls. There seems to be very little known about dissociation, and somewhat vague theories have been advanced to explain the phenomenon.

The fact is, nevertheless, apparent as shown in the production of water and other producer gases by the use of steam in contact with highly incandescent fuel. It is known that a maximum explosive mixture of pure gases, as hydrogen and oxygen or carbonic oxide and oxygen, suffers a contraction of one-third their volume by combustion to their compounds, steam or carbonic acid. In the explosive mixtures in the cylinder of a motor, however, the combining elements form so small a proportion of the contents of the cylinder that the shrinkage of their volume amounts to no more than three per cent of the cylinder volume. This by no means accounts for the great heat and pressure differences between the theoretical and actual effects.

Efficiency of Conversion of Heat to Power.—The utilization of heat in any heat engine has long been a theme of inquiry and experiment with scientists and engineers, for the purpose of obtaining the best practical conditions and construction of heat engines that would represent the highest efficiency or the nearest approach to the theoretical value of heat, as measured by empirical laws that have been derived from experimental researches relating to its ultimate volume. It is well known that the steam-

engine returns only from twelve to eighteen per cent of the power due to the heat generated by the fuel, about 25 per cent of the total heat being lost in the chimney, the only use of which is to create a draught for the fire; the balance, some 60 per cent, is lost in the exhaust and by radiation. The problem of utmost utilization of force in steam has nearly reached its limit. The efficiency of a highly refined gasoline-engine may reach 24 to 30 per cent, though the usual figures for automobile engines is 22 per cent, while the Diesel engine has shown values of over 40 per cent thermal efficiency.

The internal-combustion system of creating power is now well established in practice, and is settling into definite shape by repeated trials and modification of details, so as to give somewhat reliable data as to what may be expected from the various types when used as a prime mover. For small and large powers, the gasoline-engines are forging ahead at a rapid rate, filling the numerous wants of the engineering as well as the aeronautical world for a power that does not require expensive care, that is perfectly safe at all times under normal conditions of use, that can be used in any place in the wide world to which its concentrated fuel can be conveyed, and that has eliminated the constant handling of crude bulky fuel and water, as the steam-engine requires when fired with coal.

Requisites for Best Power Effect.—The utilization of heat in a gas-engine is mainly due to the manner in which the products entering into combustion are distributed in relation to the movement of the piston. The investigation of the foremost exponent of the theory of the explosive motor was prophetic in consideration of the later realization of the best conditions under which these motors can be made to meet the requirements of economy and practicability. As early as 1862, Beau de Rocha announced, in regard to the coming power, that four requisites were the basis of operation for economy and best effect. 1. The greatest possible cylinder volume with the least possible cooling surface. 2. The greatest possible rapidity of expansion. Hence, *high speed.* 3. The greatest possible expansion. *Long stroke.* 4. The greatest possible pressure at the commencement of expansion. *High compression.* That he was not far wrong and that he had a good understanding of the problem involved even at that early date is borne out by events that have transpired since his first conception of the basic principles he enumerated over 65 years ago.

Thermodynamics of Aircraft Engines.—In a paper entitled "The Thermodynamics of Aircraft Engines" published in the *Journal of the Royal Aeronautical Society of England,* in February, 1924, Mr. H. R. Ricardo, the well known internal-combustion engine authority covers the subject of ideal efficiency and why it cannot be completely realized in a very thorough and interesting manner. In this paper, the purpose of which was to outline the chief thermodynamic problems in connection with gasoline engines, especially those used in aircraft, Mr. Ricardo first discusses the factors that prevent the attaining of the ideal air-cycle efficiency. Following the statement that the power output and the efficiency of any internal-combustion engine depend primarily on the ratio of expansion, since this governs the proportion of the total heat supply which can be converted into useful work on the piston, it is shown that ideal efficiency pre-supposes two conditions: (*a*) that the working medium is pure dry air and (*b*) that no interchange

of heat between the air and the surrounding walls or from the previous cycle takes place. The points brought out in the discussion of the factors that hinder the realization of ideal engine efficiency are indicated below.

The increase of specific heat at high temperatures represents a dead loss as compared with the air-cycle basis, since it means in effect that any heat added at high temperatures is not accompanied by so large an increase in pressure as at the lower temperatures. At the temperatures prevailing due to combustion in an aircraft engine or indeed in any gasoline-engine, namely, between 1,800 and 2,500 deg. Cent. (3,272 and 4,532 deg. Fahr.), the loss due to the change of the specific heat is of a very large order and constitutes the greatest discrepancy between the practically possible and the air-cycle ideal. Dissociation of the products of combustion at high temperatures does not play such havoc with the power and the efficiency as the change of the specific heat, because it occurs to an appreciable extent only at very high temperatures, so that as the expansion proceeds and the temperatures fall, re-combination with a corresponding liberation of heat takes place and a part of the lost heat is recovered, in time for some, though not the full, use to be made of it.

Heat Loss to Cooling System.—Heat loss to the jackets is yet another factor that prevents the attaining of the ideal air-cycle efficiency. If in an aeronautic engine of good modern design the entire flow of heat to the jackets could be suppressed, the net gain in power and efficiency would be about eight per cent. It is clear that the above-mentioned three main sources of loss are directly dependent upon the flame temperature and that if we can reduce this temperature, we shall at once gain in efficiency. Nor is that all, for nearly 90 per cent of present-day engine troubles are due directly or indirectly to excessive heat flow: directly in the form of burnt-out valves, sparkplugs and pistons; indirectly in the form of carbonized oil, gummed-up piston-rings and defective cylinder and combustion head parts lubrication.

With reference to the interchange of heat between the residual products of the last cycle and the fresh mixture, the value of absorbing the heat of the products of the previous cycle is stressed. The influence of the latent heat of vaporization of a fuel is often overlooked, yet it is the one characteristic above all else which determines the power output available from an engine. The fuel with the highest latent-heat will give the highest power output, provided it is volatile enough. Mr. Ricardo discusses the difference in this respect between the engine working on the explosion cycle and the compression-ignition engine. Regarding the pre-burning of the mixture during the suction and the compression stroke, it is stated that it is difficult to prove convincingly the presence of this pre-burning, though there can be little doubt as to its existence; it is still more difficult to determine its extent, which is controlled clearly by both the time element and the temperature of the surface. In the ordinary engine cylinder the only surface that is likely to cause pre-burning is the head of the exhaust-valve, and the extent of pre-burning from this surface probably varies as something like the cube of its temperature.

The Combustion Process.—Taking up the question of the actual process of combustion in an engine cylinder, Mr. Ricardo says in part;

"As a mental picture the writer prefers to regard the process as though it developed in two entirely distinct stages, one the growth and the development of a self-propagating nucleus of flame and the other the distribution of this flame throughout the combustion-chamber. The former is a chemical process depending upon the nature of the fuel, upon both the temperature and the pressure and also upon what Tizard terms the temperature coefficient of the fuel, that is the relation between the flame temperature and the rate of acceleration. The second stage is a mechanical one, pure and simple. I have suggested that these two stages are separate and distinct, but they must, of course, interact upon one another to some extent, for example the higher the flame temperature and the more rapid the rate of burning, the more rapidly will combustion spread with a given rate of turbulence."

Discussing the first stage in the process of combustion, Mr. Ricardo describes at length two long series of experiments that have been carried out by Fenning at the National Physical Laboratory and by Tizard and Pye at his laboratory.

Turbulence of Great Value.—The second stage, it is stated, depends upon turbulence, without which no internal-combustion engine could run. The two essential functions of turbulence are (*a*) to spread inflammation and (*b*) to scour away the stagnant layer of mixture which, clinging closely to the cold cylinder walls, can get rid of its heat so rapidly as either to escape combustion altogether or to burn so late in the expansion stroke as to be of very little value. An increase of turbulence results in the following:

(1) The stagnant layer of gas adhering to the walls is scoured away and brought into use
(2) The rate of inflammation is hastened; but
(3) The heat loss to the walls is increased; and
(4) After a certain point there is a tendency to reduce the possible range of burning

Mr. Ricardo states further that experiments which he has carried out recently indicate that each fuel has an optimum degree of turbulence beyond which any further disturbance results merely in excessive heat-loss and a narrowing of the possible mixture-range, while he has succeeded in reaching such a degree of turbulence as to prevent ignition altogether, presumably by the dissipation of the nucleus of flame before it can become fully established at the plug points; in other words, the draft may be such as to blow out the candle. The importance is stressed of determining how far the rate of burning and detonation are dependent upon pressure alone as distinct from temperature, and Mr. Ricardo expresses his conviction that under actual working conditions, pressure and pressure alone plays a supremely important part, but it is very difficult to prove this because the temperature control at best is somewhat indeterminate.

Stratified Charge.—Reference is made to stratifying the charge, separating it into two component parts, one consisting of a comparatively rich and rapid-burning gasoline-air mixture, in the neighborhood of the sparkplug, and the other of pure air to act as a diluent; these two components being kept more or less separate until after the ignition of the first.

By such means a high initial flame temperature is obtained; but, as soon as inflammation is fairly started, the flame temperature is reduced by dilution with the excess air present in the cylinder. Experimental work in this connection is described, and the advantages of stratification are stated, especially for aircraft. The advantages and the drawbacks of supercharging are discussed, and it is stated that if one works with a stratified charge, one can then supercharge to almost any extent. In this connection the comparative suitability of a sleeve valve and a poppet valve engine were discussed and advantages of the sleeve valve were brought out.

Detonation Prevention Important.—To sum up, the biggest item of loss is that due to the change of the specific heat; next comes the direct heat-loss to the cylinder walls, followed by detonation which limits the expansion that can usefully be employed, dissociation and pre-burning. The first two are both directly dependent upon the flame temperature and the next two largely so. Anything therefore that can be done to reduce the flame temperature will be a direct gain, but with all fuels except hydrogen a very high temperature is needed to promote combustion; it would seem therefore that it is necessary to concentrate on trying to stratify the charge and by so doing try, in effect, to mislead the mixture by allowing a high flame temperature at the start but diluting it immediately afterward.

Pre-burning is no doubt a serious trouble, for not only is some of the charge lost but, when the temperature is raised prematurely, detonation is provoked, and so a lower compression must be used than would otherwise be necessary. Hot exhaust valves are the main, almost the sole, cause of pre-burning, which can be eliminated almost entirely by the use of sleeve valves, as they involve no highly heated surfaces in the combustion-chamber. Cooling the exhaust valve by use of salt in the hollow stem and head also reduces the valve head temperature.

In conclusion, Mr. Ricardo says:

"If the writer had at this moment to design engines for aircraft, he would, for the light, fast-scouting machines, go for an air-cooled radial engine with poppet valves, a very high compression, using a fuel of very high latent heat for getting off the ground, with possibly a little, but certainly only a very little, direct supercharging at high altitudes. For the heavier classes of machine he would, in the light of present knowledge, go for a water-cooled sleeve valve engine with stratified supercharge."

The first part of this now four-year-old prophecy relating to air-cooled radial engines is now properly realized and generally followed but there has been no great increase in the use of stratified supercharge and sleeve valve water-cooled engines, only one sleeve valve engine of the Knight type being produced by Panhard in France.

Diesel Engines for Aircraft.—No considerable progress has as yet been made as to the development of the Diesel engine in making it suitable for installing in airplanes. The difficulties of design undoubtedly increase as the size of the engine decreases. The principle of working is as follows:—The engine is made in three forms, viz., as a four-stroke cycle single-engine shown in diagrammatic form at Fig. 33, as a two-stroke single-acting, and as a two-stroke double-acting engine. An essential feature in the actual Diesel engine is that it requires, besides its own cylinders and pistons, an

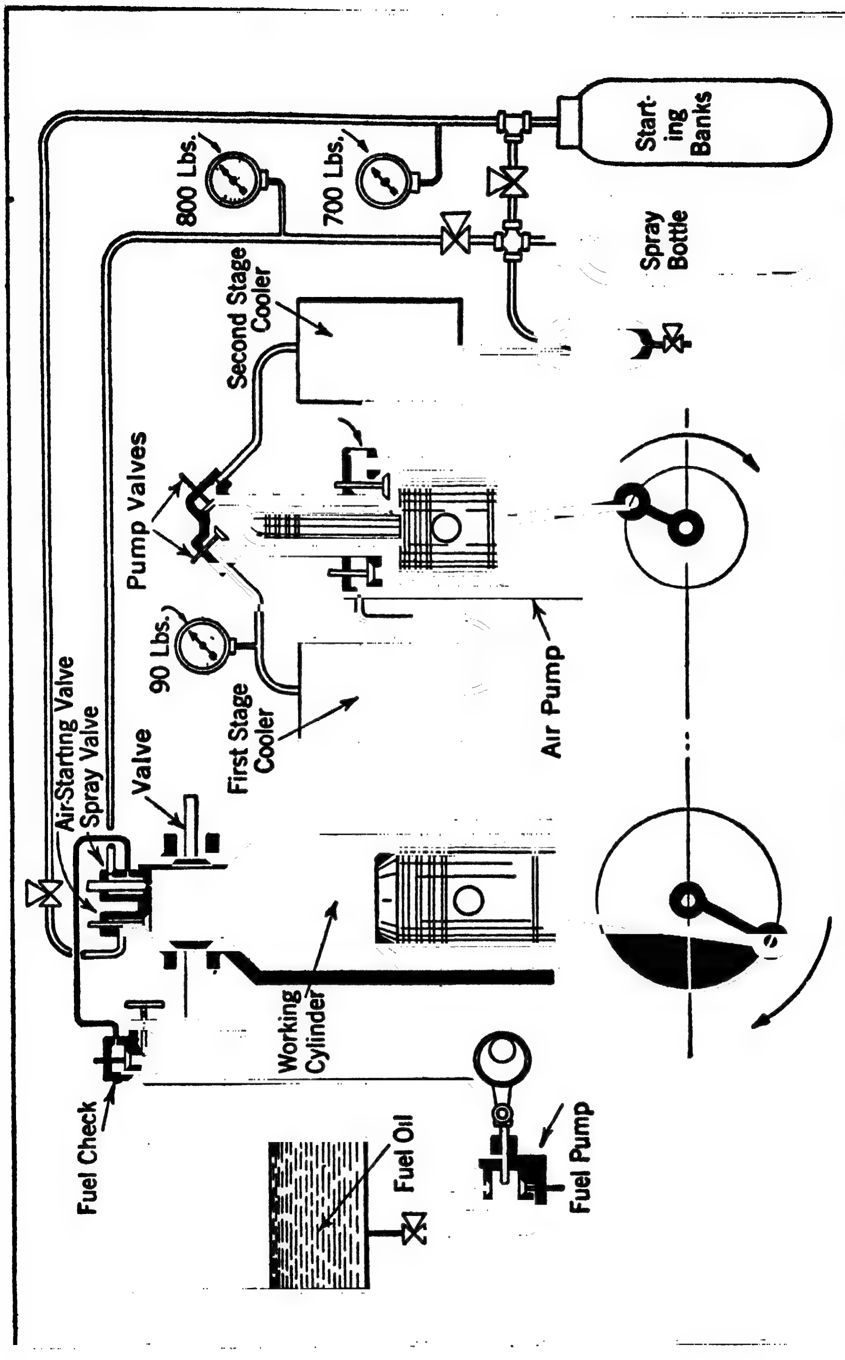


Fig. 33.—Diagrammatic Arrangement of Four-Cycle Air Injection Diesel Engine Showing Auxiliary Apparatus Required.

auxiliary air compressor capable of producing a pressure of up to 800 lbs. per square inch. The principal difference between the Diesel cycle and the ordinary gasoline motor cycle is that air only is compressed in the cylinder instead of a mixture of air and fuel.

Four-Stroke Diesel Engine Action.—In the four-stroke Diesel engine there is an air inlet valve, a fuel spray valve, and exhaust valve worked by cams, and a valve for starting purposes. The action is as follows:

First (down) Stroke.—The inlet valve is opened during the whole of the stroke, and the cylinder becomes filled with atmospheric air at atmospheric pressure.

Second Stroke.—The air valve is closed and the piston returns to the top of the cylinder compressing the air. The clearance is so proportioned that in ordinary working the pressure becomes about 500 lbs. per square inch and the temperature of the air about 1,000 degrees Fahrenheit. Near the end of the stroke a quantity of fuel is injected through the fuel valve into the cylinder by means of a blast of compressed air supplied from the auxiliary air compressor. The construction of the fuel valve is such that the oil is divided into a spray of fine particles. These, coming into contact with the very hot air, ignite automatically, and combustion takes place, increasing the pressure of the cylinder.

Third (down) Stroke.—The fuel valve remains open for about one-tenth of the stroke. During the portion of the stroke the pressure remains practically constant, about 600 lb. per square inch. During the remainder of the stroke the products of combustion expand. This stroke is the working stroke.

Fourth Stroke.—The return of the piston (during which time the exhaust valve is open) drives out the burnt gases. After this the cylinder starts afresh. An inspection of the diagram at Fig. 33, which shows the apparatus necessary for the proper functioning of a four-stroke Diesel will serve to show the reader that such engines, because of auxiliary apparatus must always be heavier than the gasoline engines unless the Diesel principle is greatly modified.

Two-Stroke Diesel Engine Action.—In some of the two-stroke engines the cylinder head is similariy fitted with fuel valve and air valve for starting purposes, and with an air valve called a scavenging valve, but the exhaust valve is replaced by ports cast in the cylinder walls. In some makes of two-stroke engine, scavenging ports are also cast in the sides of the cylinder. The cycle of operations is then as follows:—When the piston is at the bottom of the stroke, the cylinder is full of air at atmospheric pressure, which air has just been admitted through the scavenging valve. During the up stroke the air is compressed to a high pressure, and attains a high temperature. At the top of the stroke, fuel is injected by the blast of air into the hot air within the cylinder, combustion takes place, and the gases expand, thus acting on the piston. When the piston has reached about six-sevenths of the down stroke, the piston commences to uncover the exhaust ports cast in the cylinder walls. The scavenger valve opens and air is admitted under low pressure, about five pounds per square inch, which blows out the burnt gases and fills the cylinder with air ready for the

next compression stroke. Different arrangements are used by different makers for supplying the scavenging air, some makers using a separate air pump, other makers using a stepped or double piston. In the double-acting engine the piston is attached to a piston rod which works through a gland and metallic packing at the bottom of the cylinder. A cycle of operations, similar in action to that described for the two-stroke engine, takes place at both ends of the cylinder, but these types are very heavy and have been devised primarily for stationary and marine applications.

The main difficulty in applying the Diesel engine to aircraft seems to be in the design of parts that will be light and yet suffciently strong to withstand the high pressure necessary in injecting fuel against pressures much higher than ordinarily met with in automotive engines. With the high compression it is evident that gaseous mixtures cannot be used owing to pre-ignition troubles. By taking in air alone, compressing it to such a high degree where it is sufficiently hot to ignite injected fuel, it is possible to attain a higher thermal efficiency, and inasmuch as the fuel is admitted in successive small quantities, the power will endure during the greater part of the power stroke and will be due to a series of impulses, due to burning, rather than the single violent shock or high explosive pressure produced at the time of explosion in the four-cycle engine using ordinary gasoline where the mixture is seldom compressed to more than 100 lbs. per square inch prior to ignition. The mean effective pressure in the Otto cycle engine results in having the maximum pressure as close to the inception of the power stroke as possible and as a result, with the small throttle opening used in operating a plane at speeds under its normal cruising value, this means, effective pressures fall off faster than the resulting diminishment in the fuel consumption justifies. This, of course, produces relatively low thermal efficiency. This factor of engine operation at partially closed throttle is met with in automobile practice much more than in airplane engines which usually require half throttle or better to insure flight so the engines are operating under more favorable conditions to secure good thermal efficiencies.

Port-Scavenging Type.—Fig. 32 A shows a diagrammatic arrangement of a two-cycle, port-scavenging engine of the simplest type. In this system, the scavenging air, which is furnished by an air pump driven from the main crankshaft, is admitted by means of ports in the cylinder which are opposite a row of similar exhaust ports. As the piston descends, the exhaust port is first uncovered and the exhaust gases escape. As the piston further descends the row of scavenging ports is uncovered and scavenging air from the scavenging pumps blown into the cylinder. This air is at from six to eight pounds pressure and is deflected upwards by the deflector on the piston and by the angle of the scavenging ports. This air forces out most of the remaining gases and on the return of the piston the scavenging ports are again covered, after which the flow of scavenging air will stop. The piston then covers the exhaust ports and compression begins.

Overhead Valve Scavenging Type.—With this type of two-cycle engine the scavenging air is admitted by means of from one to four valves in the cylinder head as shown in the bottom series of diagrams Fig. 32 A. The action is as follows:

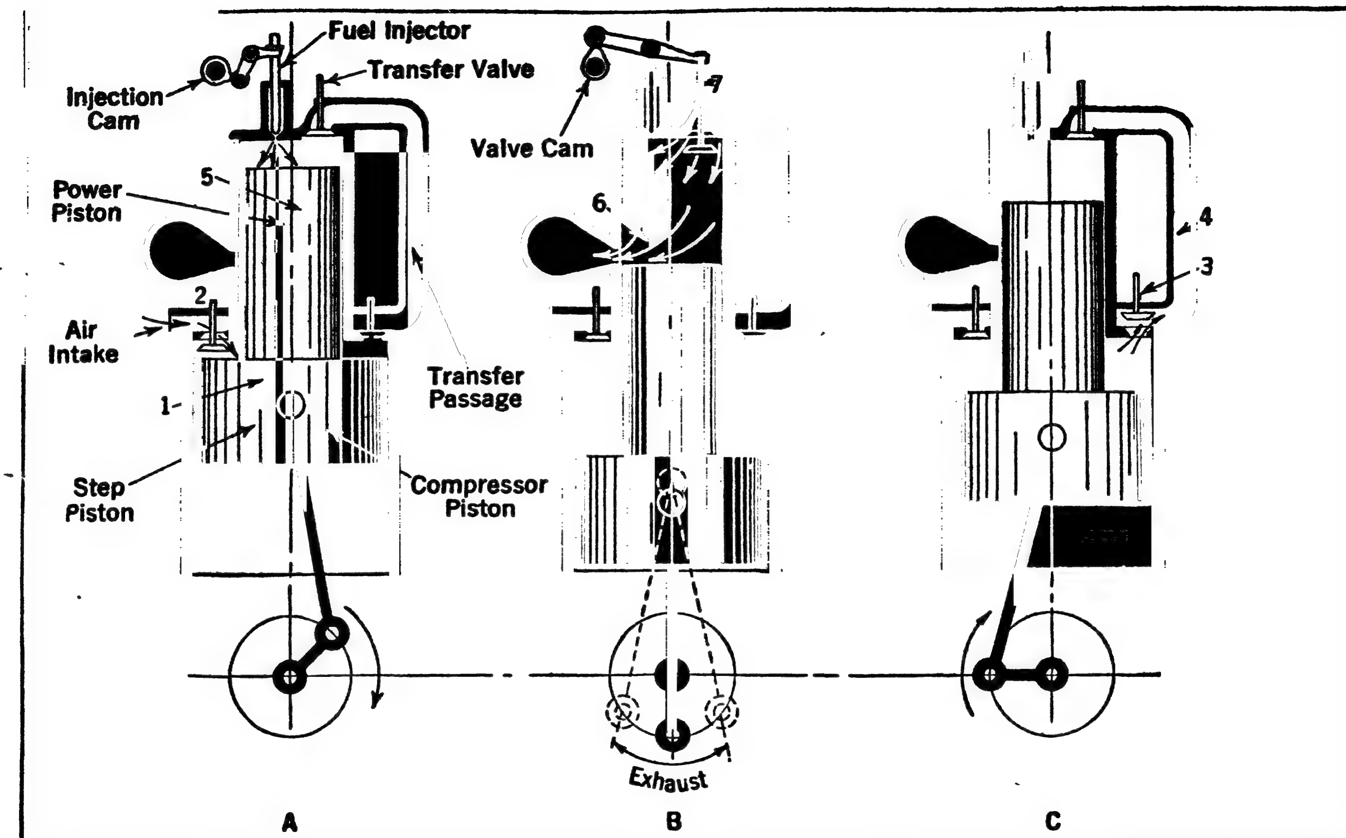


Fig. 34.—Diagrammatic Arrangement of Two-Cycle Step Piston Type Diesel Engine in which the Enlarged Portion of the Piston and the Cylinder Surrounding it Act as an Air Compressor.

(A) Fuel is being injected into the cylinder and the piston is going down on its power stroke. All valves and ports are closed with the exception of the stray valve which remains open for about 30 degrees past top center.

(B) The piston starts to uncover the exhaust ports, reducing the pressure to atmospheric.

(C) An instant later the overhead scavenging valves open, and air, furnished by a separate air pump, enters and blows out the remaining gases through the exhaust ports.

The fuel consumption of the two-cycle engine is slightly greater than the four-cycle engine, chiefly on account of the successive scavenging and charging of the latter which is more complete. The conversion of the fuel into useful work and distribution of the different heat losses of the two types of engines is shown diagrammatically in Fig. 36. The two-cycle engine has proved much more difficult to design, and the scavenging of the cylinders has to be solved mostly by costly experiment. If the scavenging is not perfect the working of the engine will prove defective. Instead of having a cylinder full of air for combustion, with imperfect scavenging there remains a quantity of burnt gases which reduces the efficiency of the engine by not properly utilizing all the fuel.

High-Speed Diesel Engines.—In discussing the subject of "High Speed Diesel Engines" before the S.A.E., O. D. Trieber, an expert on marine Diesel engines and a member of the Society gives some pertinent facts on this subject that will show why there are some difficulties inherent with true Diesel engine construction that must be overcome before the advantages can be realized in lighter, high-speed automobile and aircraft applications. Most solid-injection Diesel engines operate on a dual or mixed cycle, that is, a combination Otto-Diesel cycle; and, of late, air-injection engines have been built for a similar dual cycle. In actual practice the dual cycle can be carried to a point that, within reasonable constructional limits, will show fuel economy of about 0.35 lb. per b.hp.-hr., which accounts for the tendency of air-injection engines toward the dual cycle.

Diesel engines were first made to burn coal-dust, the compression-pressure being about 900 lb. per sq. in. Later, the perfecting of air injection brought the compression-pressure down to 500 lb. per sq. in., and now certain types of solid-injection Diesels start from a cold condition with the compression-pressure at 325 lb. per sq. in. It is interesting to note that the compression-pressure of carbureting engines has been gradually raised. Diesels and carbureting engines, though radically different in type, are, in Mr. Trieber's opinion gradually approaching each other. Furthermore, carburetion has been improved to such an extent that the higher grades of fuel-oil necessary for some Diesel oil-engines at present on the market can be, and are being, used in carbureting engines with electric ignition and with marvelous economy. These developments naturally militate against the development of self-ignition Diesel engines.

Marine Diesel Engines Very Heavy.—The Diesel engine, as originally developed, was very large and heavy, and followed the generally recognized practice in the art of building large marine-engines, in which slow speed was necessary for propeller efficiency. The marine field for years offered

the only Diesel-engine market; in fact, if it were not for marine business, there would be little or no activity in the building of Diesel engines in America today. For years Diesels have been built that weigh from 175 to 500 lb. per hp. Slow speed was the rule. No market for fast-running engines existed. Attempts were made from time to time to develop higher speed and obtain lighter weight, particularly in small engines, but what little success was achieved was quickly discounted by the further development of carbureting engines using electric ignition. The whole world was organized to provide a carbureting fuel-oil at every turn of the road, chemists introduced the cracking process to produce more carbureting fuel from the crude, and now synthetic fuels doped for terrifically high compressions and using electric ignition are approaching Diesel-engine economy very closely on a weight-per-horsepower basis to the order of 0.5 to 0.4 lb. of fuel per b.hp.-hr. Furthermore, the world has been organized to produce carbureting engines at very low first-cost and to service them, a general knowledge of their peculiarities has been increasing, and supplies of spare parts are well distributed throughout the country. A new era in Diesel engineering has arrived. Engineers are taking advantage of the progress made in the carbureting engines used in automobiles and airplanes and are using the new metal alloys now available. It is natural, therefore, that progress should be expected in reducing the weight of the larger engines, insofar as the market will absorb the product and provide opportunities for further advance.

The principle used in Diesel engines is becoming generally recognized as the ignition of fuel by the heat of compression in internal-combustion engines. In the Diesel field a battle has been waged between the advocates of air injection and solid injection, and of self-ignition and semi-Diesel or semi-self-ignition of both air and solid injections. The systems all have served well, but Mr. Trieber believes he is safe in saying that air injection and solid injection have held the limelight for some years, with solid injection now gaining undisputed favor in every field. The Diesel cycle has been well established as a constant-pressure cycle, as compared with the constant-volume, or Otto, cycle.

Diesel Mean Effective Pressure Low.—The mean effective gas-pressures obtainable in a Diesel engine using natural aspiration are inherently lower than those of a carbureting engine, because the expansion ratio is reduced by the use of the dual cycle. The subject of mean effective pressure in Diesel engines has been much contested for years among Diesel-engine builders. Attempts have been made to establish a standard of mean effective pressure. The fact of the matter is that an air-injection Diesel engine using natural aspiration and a true Diesel-cycle has a limiting brake mean effective pressure of about 80 lb. per sq. in., beyond which it is difficult to go. Some of the semi-Diesels have been limited to 45 and 50 lb. per sq. in. b.m.e.p. A carbureting engine today easily develops 110 lb. per sq. in. b.m.e.p., and some as high as 140 lb. per sq. in., with even higher records. So, the air-injection Diesel seems to have a further handicap in weight per horsepower of ratios from 80 to 110, and up to 140.

For several years Mr. Trieber has endeavored to increase the mean effective pressure in Diesel engines. Ten years ago, to get 75 lb. per. sq.

in. was an effort. A study of the theoretical limits of the mean indicated pressure, when the dual cycle is used, offers encouragement. An analysis has been made with the following assumptions: 350 lb. per sq. in. gauge adiabatic compression-pressure with an exponent of 1.35, which is comparable with engine performance; 100 per cent volumetric-efficiency, the fuel charge to be chemically correct for complete combustion, no excess air, and sufficient oil to be burned at constant volume to raise the pressure at top dead-center to 700 lb. per sq. in., and the combustion to be continued at constant pressure until it is complete; the expansion to be adiabatic, using the mean gas-constant as exponent, which shows a theoretical mean indicated pressure of 245 lb. per sq. in., which we can approach but never reach, as there are losses due to radiation, dissociation and excess air.

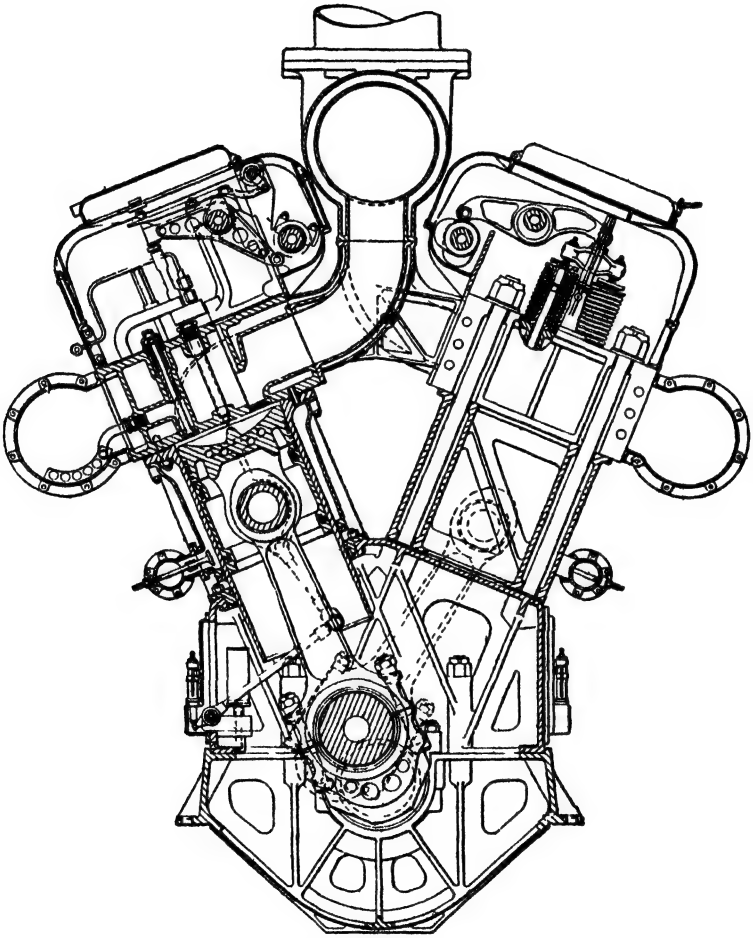
The theoretical indicated thermal efficiency is 45.6 per cent, corresponding to a fuel consumption of 0.308 lb. per i.hp.-hr.; at 84 per cent mechanical efficiency, the fuel consumption is 0.366 lb. per b.hp.-hr. If the above figure of 245 lb. for the theoretical limit of the mean indicated pressure is corrected for the losses we know to exist; namely, radiation, twenty per cent; excess of air, ten per cent; volumetric-efficiency loss, ten per cent, we must first correct for volumetric efficiency, namely, $245.00 \times 0.90 = 220.50$ lb. Then, correcting for the excess air required, we have $220.50 \times 0.90 = 198.45$ lb. Many engineers, particularly European engineers, consider that ten per cent excess of air is not enough. Carbureting engines, however, have probably exceeded this amount. By correcting the foregoing figure for losses by radiation, we have $198.45 \times 0.80 = 158.70$ lb.

Correcting the theoretical indicated thermal efficiency of 45.6 per cent, which is equivalent to 0.308 lb. of fuel per i.hp.-hr., and using fuel with the low heat-value of 18,500 B.t.u., we have the correction for radiation losses, twenty per cent, and for mechanical losses, sixteen per cent, bringing the fuel consumption to 0.43 lb. per b.hp.-hr.

Liquid Fuel Atomization a Problem.—In a Diesel engine, a low volatile fuel-oil must be converted from a cold liquid-condition into a finely divided or atomized state and its temperature must be raised to a point at which the hydrocarbon atoms unite chemically with the oxygen in the air charge. This is the crux of many limitations. We do not know how to convert the fuel successfully into such a state before it is injected into the air charge. The greatest inherent difficulty in doing so is the decomposing of liquid fuel-oil while being converted into a gaseous state. If we find a way to convert heavy liquid fuel-oil into a gas without breaking it down, decomposing it, or freeing the carbon, we shall have a solution to the burning of oil in cylinders that will put the Diesel engines of today into the discard; but the nature of the oil cannot be changed. Time is required to convert a liquid fuel into a finely divided or atomized state, raise its temperature sufficiently to unite it chemically with the oxygen, and produce combustion. This time is called the time-lag. The injection cannot be admitted too early in the cycle, for the fuel would enter the air at too low a temperature to produce ignition. By the time the heat from the compression became sufficient to ignite the hydrocarbons, so great a quantity of fuel would be in the air charge, which would be rising in temperature at the same time and rate, that, when the temperature was sufficient to produce a chemical

union of the hydrocarbons and the oxygen, the action would be of such volume or quantity that the cylinder pressures would be raised above the practical limits.

In practice, as is to be expected, Mr. Trieber finds that the fuel consumption is reduced as the mean indicated pressure is reduced, and the amount of fuel burned at constant volume is maintained constant; in other words, by reducing the "lap" and maintaining the "lead" of the fuel spray, we have records of economy as low as 0.34 lb. per b.hp.-hr., but the brake mean effective pressure was low, being about 80 lb. per sq. in. Thus, we see that it is possible to obtain mean effective pressures equal to those of



—CROSS-SECTION OF 3000-HP. DIESEL ENGINE

Fig. 35A.—Cross Section of a 3,000 Horsepower Marine Diesel Engine Operating at 700 R.P.M., Weighing 58,000 Pounds, Presented to Show Massiveness of Construction Required in Heavy Duty Engines of this Type.

the carbureting engine, but the higher initial pressures will always make it necessary for the Diesel engine to be inherently heavier, with heavier reciprocating parts; and this again is a barrier to speed. The speed is limited for two reasons; the first, mechanical, as noted, and the second, more important in small engines, by the time-lag of fuel ignition. Besides this, the introduction of relatively small quantities of fuel by an injection valve is a delicate process more difficult to realize in its practical application to small engines than theoretical considerations would indicate.

Aircraft Diesel Engines Not Yet Available.—It is believed that if the constant pressure engine can be refined and developed to give high efficiency at r.p.m. comparable to average airplane cruising speeds that it will be possible to make a material reduction in the cost of operation because the cheaper, nonvolatile fuels can be used instead of the more expensive liquids that evaporate readily and which are absolutely necessary where liquid vapor is drawn into the cylinder and compressed before ignition. It is believed that in the light of present-day developments, such as the knowledge that now obtains relative to high grade steels and other metals and alloys of great strength and lightness, that constant pressure cycle engines operating approximately on the Diesel principles will be developed that will not be impractical on some types of commercial aircraft. The Diesel engine is extremely attractive for aircraft from two viewpoints; first reduction of fire hazard and, second, increasing the cruising range of aircraft; in fact, after the phenomenal long-distance records of 1927, experts prophesy that there will be no great advance in that direction until we have Diesel aircraft engines, which promise to reduce the fuel consumption and eliminate all troubles due to electric ignition now essential with conventional engines. They may, however, introduce other troubles of a mechanical nature due to fuel injection system.

Hon. Edward P. Warner, Ass't Sec'y of the Navy for Aeronautics, said in a recent discussion of the subject: "It has occasionally been suggested that the Government departments, especially the Navy, have been indifferent to the development of Diesel powerplants. I repudiate any such suggestion. We are fully appreciative of the great theoretical possibilities of the Diesel, especially on long flights of airplanes or airships where fuel consumption becomes of greater relative importance than the weight of the bare engine. I am confident that the problems of the heavy-oil compression-ignition engine for aerial service will be solved, and what we hear is most encouraging as indicating that we are on the road; but we are unable to feel that the solution, in form of an aeronautic engine ready for operation, is immediately at hand. We shall do all that we can, consistent with proper attention to those standard types of gasoline engines on which our major dependence must certainly be placed for a number of years yet, to speed its coming. When it is here it will be more than welcome, and I hope that it will ultimately bring not only decreased fuel consumption and reduced fire hazards, but increased reliability and life as well, in its train."

The Attendu Solid Injection Oil Engine.—A two-cylinder experimental engine designed by André Attendu, a French engineer and a member of

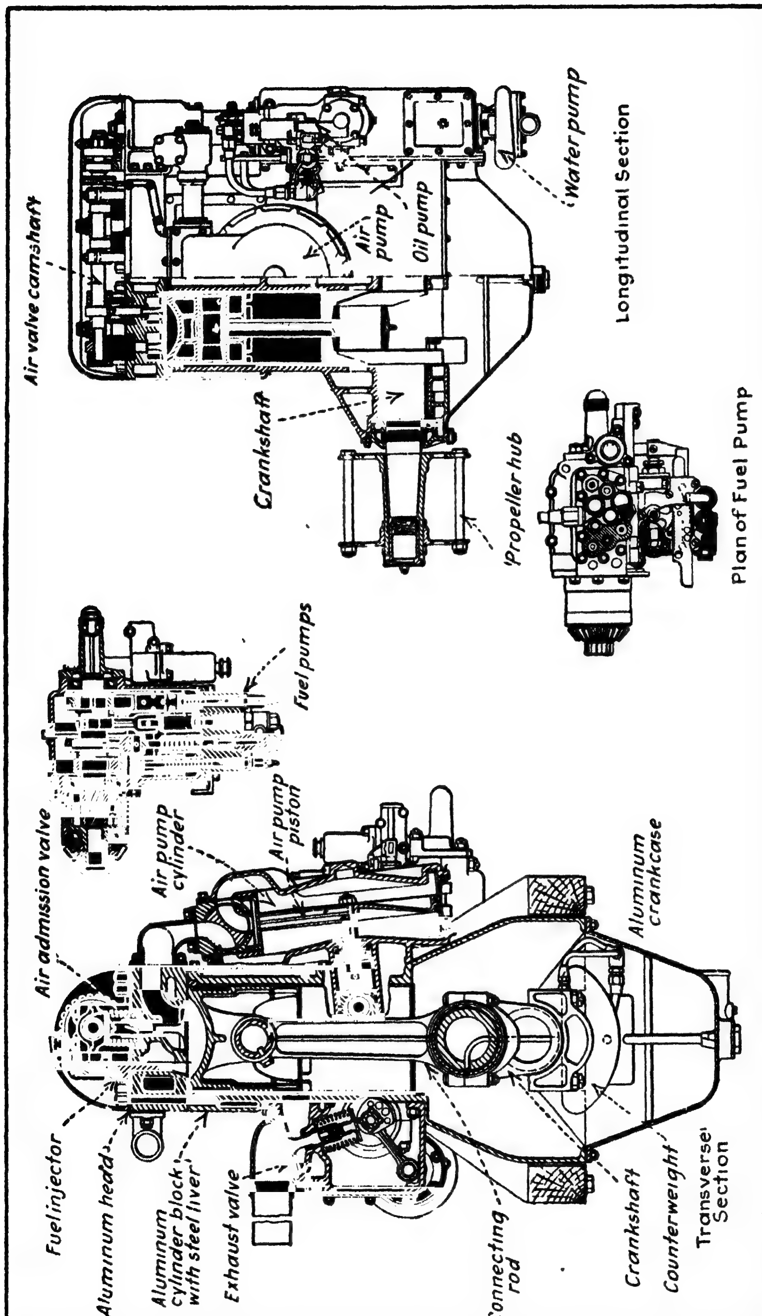


Fig. 35B.—Diagrams Showing Construction of Attendu Heavy Oil Aviation Engine Built for U. S. Navy Experimental Work which has a Remarkably Low Weight per Horsepower for this Type of Construction.

the S. A. E., for the United States Navy is shown at Fig. 35 B, and tests that have been made to date as reported in the *S. A. E. Journal* have given very promising results. It was built primarily to determine if a light weight engine could operate at high speeds under the high pressures that prevail in oil engines.

This aviation engine, which is the most recent and advanced engine built by Mr. Attendu is of the high-compression self-ignition type. It operates on the two-stroke cycle, using solid, that is, airless, injection and can be started from cold on its normal fuel, which is 0.93-specific-gravity fuel oil. The engine is extremely flexible; it is capable of maintaining a high torque throughout a speed range of from 400 to 1,600 r.p.m. This flexibility is obtained by refinement in the regulation of the fuel pump and by a patented system of compression-pressure control. This engine has two cylinders, 5½ x 6½ inches, and a rated output of 100 b.hp. at 1,500 r.p.m. Port scavenging and uni-directional flow of air and gases are obtained by placing the inlet valves in the head. A large-diameter short-stroke air pump is mounted at the side of the engine, with its axis nearly horizontal. It is double-acting and is driven at crankshaft speed by a layshaft that also operates the exhaust valves. These valves, two per cylinder, are set in a pocket close to the exhaust ports and their time of closing is governed by an automatic control. The valves are always open when the piston uncovers the exhaust ports, but are closed at a variable point before the ports are covered again. By variation in this point of closing, the effective length of the compression stroke is altered. Thus, in starting and at lower speeds, a greater volume of air is retained in the cylinder to compensate for the slow compression and consequently greater losses of heat and pressure. Thus the final compression-pressure may be held at a sensibly uniform value. This is an essential feature of the Attendu engine and is shown in the transverse section in Fig. 35 B.

Elements of the Attendu Fuel System.—The fuel-injection system comprises three main elements: (*a*) a primary, or low-pressure, pump; (*b*) a high-pressure pump that meters and injects the fuel and (*c*) a spray nozzle or injection valve. Both the primary and injection pumps are of the single-acting plunger type and are operated from a common shaft, on which are two primary plungers driven by eccentrics. These supply two injection plungers, one per cylinder, which are cam-operated. The low-pressure stage is required to draw fuel oil from the tank and to ensure the rapid and complete filling of the high-pressure cylinder. Variation in the power output of the engine is obtained by controlling the quantity of fuel injected at each stroke and this is accomplished by lifting or lowering the high-pressure plungers in relation to the cams. This action has the effect of altering the point at which injection commences, but this is automatically compensated for by the timing mechanism of the pump. In addition, a wide range of timing control is available.

The injector consists essentially of a nozzle that is controlled by a spring-loaded needle-valve. This valve is set to retain its seat against the pressure due to the primary stage of the fuel pump, but opens promptly upon the marked increase in pressure due to the operation of the high-pressure plunger. The timing is arranged so that sensibly constant-volume

combustion is obtained at the lower speeds. As the speed is increased the cycle changes from constant-volume to constant-pressure cycle.

Results of Navy Tests.—The engine was delivered to the Navy in February, 1925, and ran fairly well up to 1,800 r.p.m., but lubrication and minor mechanical troubles developed in the valve adjustment, couplings and elsewhere, which delayed the official tests until the end of November, 1925, when the first test was passed successfully and the title to the engine vested in the United States Government.

When first delivered, the engine developed 61 b.hp. at 1,350 r.p.m. With improvements on the adjustment and especially on the lubricating oil system, the brake-horsepower increased to 76 at 1,360 r.p.m., 82 at 1,610 r.p.m., and 85 at 1,620 r.p.m. The best power output, 91 b.hp. at 1,525 r.p.m., was obtained in Mr. Attendu's laboratory and he stated that by making some other alterations that were in course of execution an additional output of from 20 to 25 b.hp. can be obtained, which will bring the engine up to between 110 and 116 b.hp. for a total weight of 417 pounds, or a dry weight of 3.6 pounds per b.hp. The fuel consumption is now in the neighborhood of 0.6 pounds per b.hp.-hr., and the expectation is to reduce it to 0.5 pounds. It shows no marked advantage in fuel consumption at present over carbureting engines as far as quantity needed is concerned but the cost of fuel would be cut to one-third or even less that amount of an equivalent quantity of aviation gasoline. It is the most promising aircraft Diesel engine that has been described and illustrated publicly up to the date of preparing this volume. The maximum speed recorded with this engine is 2,210 r.p.m. A very complete description of this engine can be found in the *S. A. E. Journal* for February, 1926.

Deutz High-Speed Diesel Engine.—The opportunity of another review of developments in the line of high-speed relatively light automotive Diesel engines in Germany was offered by the recent Berlin motor vehicle show. Several new designs were presented in addition to those already known, such as the Daimler-Benz, M.A.N., Junkers and others. Descriptions of the engines were given in *Der Motorwagen* and *Auto-Technik,* and the following information is based on articles in these publications. The illustrations are reproduced from *Automotive Industries.* The Deutz engine is similar to the Daimler-Benz, in having an antechamber or an ignition chamber. It differs from most other engines of this type in having the ignition chamber located in the cylinder head to one side, instead of in the center. A central ignition chamber limits the size of valves which can be accommodated in the head, and the reason for the use of the offset chamber is obviously to remove this limitation. The ignition chamber inset is of thimble form and has a number of spray openings in its lower end, so arranged as to direct the spray at a considerable angle to the piston head. A hot-wire igniter, used for starting, screws into the side of the cylinder head as shown at Fig. 35 C.

The Deutz engine is being built in four- and six-cylinder types, with a bore of 4.53 inches and a stroke of 6.69 inches. The four-cylinder engine develops 55 horsepower at 1,250 r.p.m. and the six-cylinder 85 horsepower at the same speed. The four-cylinder engine, inclusive of electric starter and generator, weighs about 1,320 pounds or 24 pounds per horsepower,

while the six-cylinder engine weighs 1,630 pounds or about nineteen pounds per horsepower. The engine is intended principally for marine and industrial purposes, but can also be used for tractors and trucks. It is much too heavy for aircraft use, but the principles of operation are of interest as they indicate some of the points that must be considered by designers of such engines intended for aircraft. The valves are operated from the camshaft in the crankcase through side rods and rocker levers. Camshaft drive is through gears with helical teeth which are located just inside the flywheel.

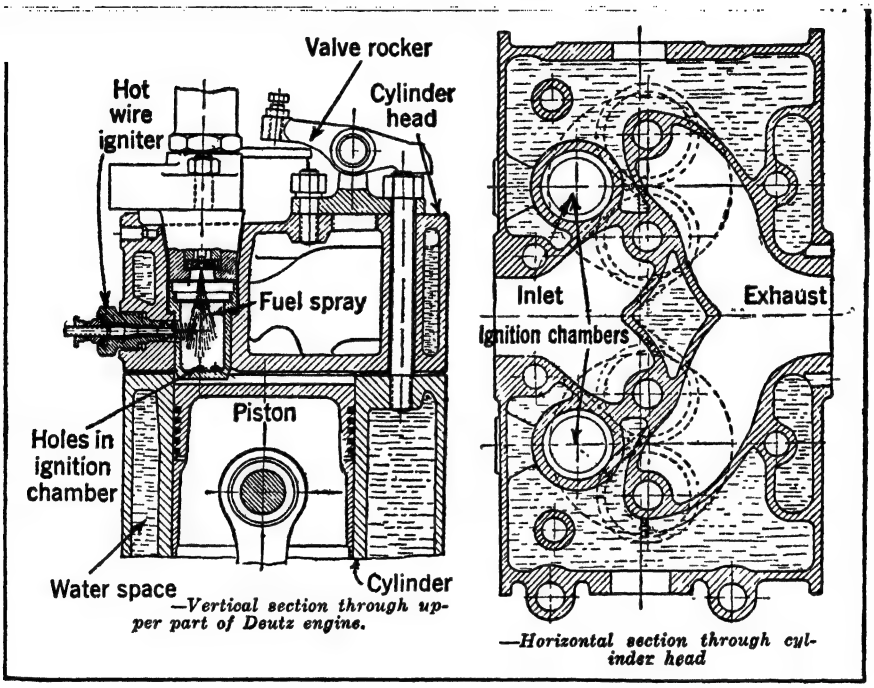


Fig. 35C.—Diagrams Showing Construction of Deutz Diesel Engine.

This arrangement of the camshaft drive gears is not very popular because of their comparative inaccessibility, but it is used here because the nodal point of the crankshaft with respect to torsional vibration is located a short distance ahead of the flywheel, and by placing the gears in this position they are made to operate more silently and their life is added to.

Deutz Fuel Injection System.—The fuel injection is regulated by means of the inclined, slidable cams which operate the fuel pump plungers through the intermediary of rocker levers. Longitudinal and cross sections through the pump are shown herewith. The cams for all of the four pump plungers are made in a single piece which is mounted on a center that can be slid along its shaft on a feather key. Presumably the roller followers are made barrel-shaped instead of cylindrical, as else there would be line contact only. The fuel pump plungers are moved outward by heavy coiled springs, to perform the suction stroke and are then returned by the cams for the delivery stroke. No means are provided for adjusting the delivery of the individual

pumps, but in their manufacture the pumps are checked so that the difference between the minimum and maximum delivery does not exceed five per cent of the delivery when the engine is idling. Means are provided whereby the time of starting the injection can be varied by hand. The fuel pump construction is shown at Fig. 35 D.

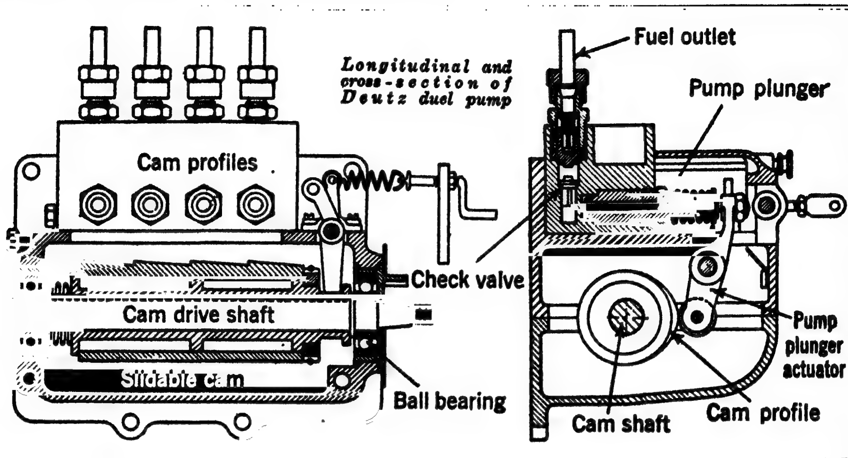


Fig. 35D.—Diagrams Showing Construction of Deutz Diesel Engine Fuel Pump.

A novel feature of the engine consists in the use of a device similar to an impulse starter as used on heavy gasoline engines, particularly tractor engines. This is inserted in the drive between the engine and the fuel pump. When the engine turns over slowly, as in cranking, a spring is

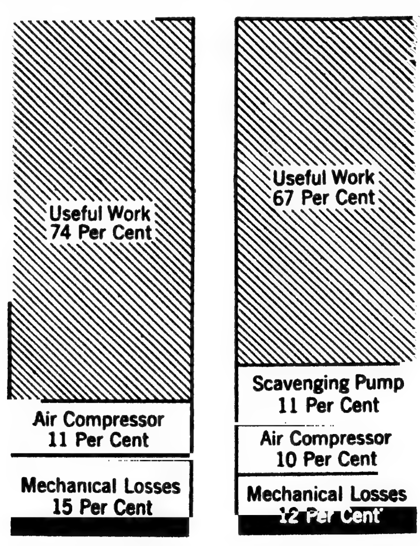


Fig. 36.—Diagrams Showing Amount of Useful Work and Distribution of Heat Losses on Two- and Four-Cycle Diesel Engine Types Intended for Stationary and Marine Use.

wound up, and at the proper time for fuel injection this spring is released, and causes the fuel pump to operate faster, and consequently to produce a higher pressure, than if it were positively connected to the engine. The higher injection pressure at low speed is desirable because it produces finer atomization.

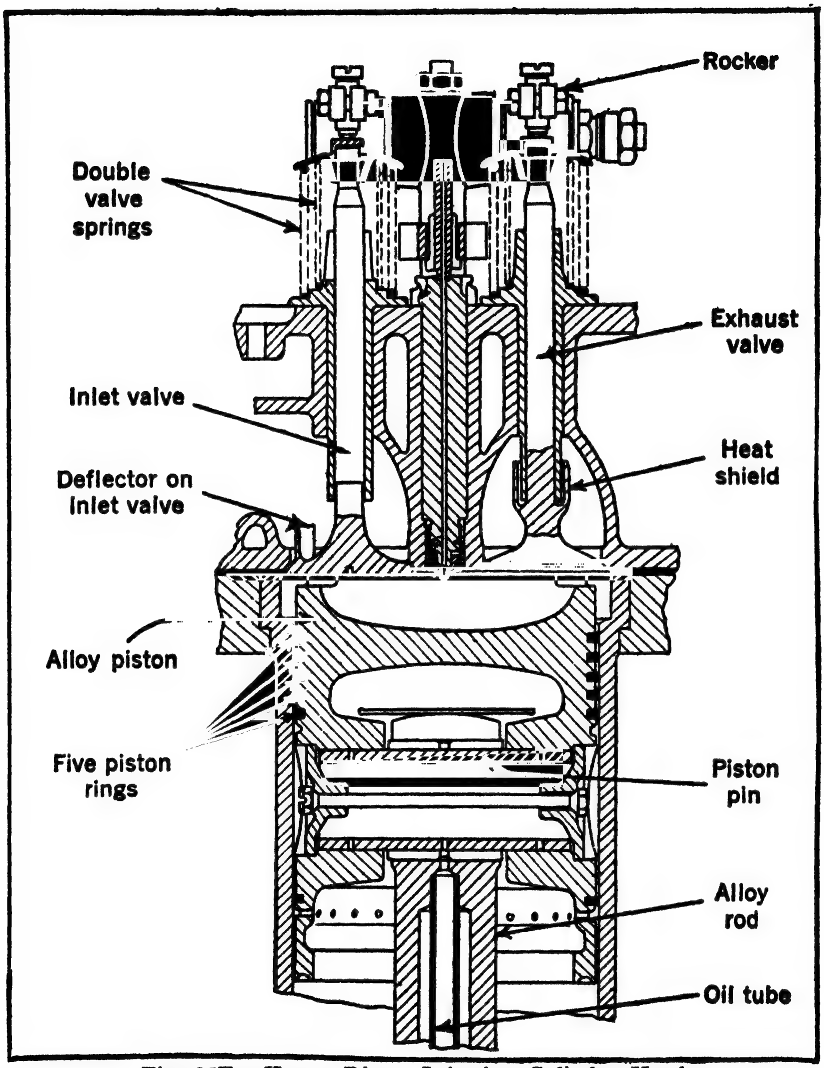


Fig. 35E.—Krupp Direct Injection Cylinder Head.

Starting is effected by means of an electric starter, and to reduce the starting torque required a handwheel is operated previous to switching on the starting current, which shifts the camshaft in such a way as to bring auxiliary cams into operation which prevent compression in the engine, and which also shift the fuel pump cams into the no-delivery position. After the engine has attained some speed, the handwheel is turned in the opposite

direction, which has the effect of placing one cylinder after another under compression and simultaneously starting the corresponding fuel pumps. If the engine is already warm, it is not necessary to "decompress" and to use the auxiliary hot wire igniter, but merely to press on the starter button.

Krupp Engine Uses Direct Injection.—The Krupp engine, shown at Fig. 35 E, which also is made in four- and six-cylinder types, uses direct injection. It has a bore of 5.32 inches and a stroke of 7.87 inches, and the horsepower ratings are 65 and 100 respectively. Light alloys are extensively used in the design, both the pistons and connecting rods being made of them. The fuel injection nozzle is of the open type and is located in the cylinder head between the two poppet valves. Cylinder heads are cast in one block for two cylinders. An interesting feature is a heat protecting sleeve for the exhaust valve guide, as shown in the drawing. The inlet valve is provided with a deflector on the bottom of its head, which is designed to increase the turbulence. These engines are being manufactured in the Krupp Germania Works in Kiel which has had extensive experience in building Diesel engines for submarines.

QUESTIONS FOR REVIEW

1. How is heat energy converted into work?
2. What is the efficiency of the conversion in the gasoline engine; in the Diesel engine?
3. Why is the internal-combustion engine more efficient than a steam powerplant?
4. Outline requisites for best power effect in internal-combustion motors.
5. Name principal types of Diesel engines.
6. Why are true air-injection Diesel engines unsuited for aircraft?
7. Explain action of step-piston Diesel engine.
8. Why is Diesel mean effective pressure low?
9. How does the Attendu engine differ from the true Diesel?
10. Why are marine Diesel engines unsuited for aircraft?

CHAPTER IV

EFFICIENCY OF INTERNAL-COMBUSTION ENGINES

Various Measures of Efficiency—Temperatures and Pressures—Factors Governing Economy—Losses in Wall Cooling—Improving Engine Performance—Effect of Increasing Compression Ratio—Use of Long Expansion Stroke—Value of Indicator Cards—Value of Compression in Explosive Motors—Factors Limiting Compression—Chart for Determining Compression Pressures—Causes of Heat Loss in Motors—Combustion Chamber Form Important—Heat Losses to Cooling Water—Horsepower Increase by Higher Rotative Speeds—Factors Limiting Aero Engine Speed.

Efficiencies are worked out through intricate formulas for a variety of theoretical and unknown conditions of combustion in the cylinder: ratios of clearance and cylinder volume, and the uncertain condition of the products of combustion left from the last impulse and the wall temperature. But they are of but little value, except as a mathematical inquiry as to possibilities. The real commercial efficiency of a gas- or gasoline-engine depends upon the volume of gas or liquid at some assigned cost, required per actual brake-horsepower per hour, in which an indicator card should show that the mechanical action of the valve gear and ignition was as perfect as practicable, and that the ratio of clearance, space, and cylinder volume gave a satisfactory terminal pressure and compression: i.e., the difference between the power figured from the indicator card and the brake power being the friction loss of the engine.

Efficiency Factors.—In four-cycle motors of the compression type, the efficiencies are greatly advanced by augmenting compression, producing a more complete infusion of the mixture of gas or vapor and air, quicker firing, and far greater pressure than is possible with the two-cycle type previously described. In the practical operation of the internal-combustion engine during the past thirty years, the gas-consumption efficiencies per indicated horsepower have gradually risen from seventeen per cent to a maximum of 46 per cent of the theoretical heat, and this has been done chiefly through a decreased combustion chamber and increased compression—the compression having gradually increased in practice from 30 lbs. per square inch to above 100; but there seems to be a limit to compression, as the efficiency ratio decreases with greater increase in compression. It has been shown that an ideal theoretical or computed efficiency of 33 per cent for 38 lbs. compression will increase to 40 per cent for 66 lbs., and 43 per cent for 88 lbs. compression. On the other hand, greater compression means greater explosive pressure and greater strain on the engine structure, which will probably retain in future practice the compression between the limits of 90 to 100 lbs. except in super-compression engines intended for high altitude work where compression pressures as high as 125 pounds have been used.

In early experiments made by Dugald Clerk, in England, with a combustion chamber equal to 0.6 of the space swept by the piston, with a compression of 38 lbs., the consumption of gas was 24 cubic feet per indi-

cated horsepower per hour. With 0.4 compression space and 61 lbs. compression, the consumption of gas was twenty cubic feet per indicated horsepower per hour; and with 0.34 compression space and 87 lbs. compression, the consumption of gas fell to 14.8 cubic feet per indicated horsepower per hour—the actual efficiencies being respectively 17, 21, and 25 per cent. This was with a Crossley four-cycle engine running on producer gas.

Various Measures of Efficiency.—The efficiencies in regard to power in a heat engine may be divided into four kinds, as follows:

I. The first is known as the *maximum theoretical efficiency* of a perfect engine (represented by the lines in the indicator diagram). This shows the work of a perfect cycle in an engine working between the received temperature + absolute temperature (T_1) and the initial atmospheric temperature + absolute temperature (T_0).

II. The second is the *actual heat efficiency*, or the ratio of the heat turned into work to the total heat received by the engine. It expresses the *indicated horsepower.*

III. The third is the ratio between the second or *actual heat efficiency* and the first or *maximum theoretical efficiency* of a perfect cycle. It represents the greatest possible utilization of the power of heat in an internal-combustion engine.

IV. The fourth is the *mechanical efficiency*. This is the ratio between the actual horsepower delivered by the engine through a dynamometer or measured by a brake (brake-horsepower), and the indicated horsepower. The difference between the two is the power lost by engine friction. It is customary to speak only of mechanical efficiency at full load, and this may vary from 80 to 90 per cent. Of the gross mechanical losses piston and piston ring friction accounts for about 60 per cent of the total. Pumping losses account for about 25 per cent and friction of bearings and auxiliary equipment accounts for about fifteen per cent.

In regard to the general heat efficiency of the materials of power in explosive engines, we find that with good illuminating gas the practical efficiency varies from 25 to 35 per cent; kerosene-motors, 20 to 30; gasoline-motors, 20 to 32; acetylene, 25 to 35; alcohol, 20 to 30 per cent of their heat value. The great variation is no doubt due to imperfect mixtures and variable conditions of the old and new charge in the cylinder; differences in engine design and accuracy of workmanship; uncertainty as to leakage and the perfection of combustion. In the Diesel motors operating under high pressure, up to nearly 500 pounds, an efficiency of 36 per cent and even more is claimed. The graphic diagram at Fig. 31 in the preceding chapter is of special value as it shows clearly how the heat produced by charge combustion is expended in an aircraft engine of average design. Some will have a different distribution of heat losses than others but these will not vary materially.

On general principles the greater difference between the heat of combustion and the heat at exhaust is the relative measure of the heat turned into work, which represents the degree of efficiency without loss during expansion. The mathematical formulas appertaining to the computation of the element of heat and its work in an explosive engine are in a large measure dependent upon assumed values, as the conditions of the heat of

combustion are made uncertain by the mixing of the fresh charge with the products of a previous combustion, and by absorption, radiation, and leakage. The computation of the temperature from the observed pressure may be made as before explained, but for compression-engines the needed starting-points for computation are very uncertain, and can only be approximated from the exact measure and value of the elements of combustion in a cylinder charge. In the light of present knowledge the highest thermal efficiency is obtained in gas- or gasoline-engines when the cylinder is between three and six inches in diameter. The highest thermal efficiency ever

TABLE III

Gas-Engine Clearance Ratios, Approximate Compression, Temperatures of Explosion and Explosive Pressures with a Mixture of Gas of 600 Heat Units per Cubic Foot and Mixture of Gas 1 to 6 of Air

Clearance Per Cent. of Piston Volume	Ratio $\frac{V}{Vc} = \frac{P + C \text{ Vol.}}{\text{Clearance}}$	Approximate Compression from 13 Pounds Absolute	Approximate Gauge Pressure	Absolute Temperature of Compression from 560 Deg. Fahrenheit in Cylinder	Absolute Temperature of Explosion. Gas, 1 part; Air, 6 parts	Approximate Explosion Pressure Absolute	Approximate Gauge Pressure	Approximate Temperature of Explosion, Fahrenheit
1	2	3	4	5	6	7	8	9
		Lbs.		Deg.	Deg.	Lbs.	Lbs.	Deg.
.50	3.	57.	42.	822.	2488	169	144	2027
.444	3.25	65.	50.	846.	2568	197	182	2107
.40	3.50	70.	55.	868.	2638	212	197	2177
.363	3.75	77.	62.	889.	2701	234	219	2240
.333	4.	84.	69.	910.	2751	254	239	2290
.285	4.50	102.	88.	955.	2842	303	288	2381
.25	5.	114.	99.	983.	2901	336	321	2440

recorded by any heat engine, namely 39.5 per cent on the net shaft horsepower, was obtained on the Napier racing aeronautic engine which won the Schneider Cup trophy in 1927. The highest thermal efficiency ever recorded on a gas-engine using town gas was obtained by A. F. Burstall, at Cambridge University, on a high-speed variable-compression engine of only 4½-in. bore; while almost, if not quite, the highest thermal efficiency ever recorded in a Diesel engine, namely 38.8 per cent on the net shaft horsepower, was obtained by the Royal Aircraft Establishment on a high-speed Diesel engine of eight-inch bore running at 1,000 r.p.m.

Temperatures and Pressures.—Owing to the decrease from atmospheric pressure in the indrawing charge of the cylinder, caused by valve and frictional obstruction, the compression seldom starts above thirteen pounds absolute, especially in high-speed engines. Col. 3 in the preceding table represents the approximate absolute compression pressure for the clearance percentage and ratio in Cols. 1 and 2, while Col. 4 indicates the gauge pres-

sure from the atmospheric line. The temperatures in Col. 5 are due to the compression in Col. 3 from an assumed temperature of 560° F. in the mixture of the fresh charge of six air to one gas with the products of combustion left in the clearance chamber from the exhaust stroke of a medium-speed motor. This temperature is subject to considerable variation from the difference in the heat-unit power of the gases and vapors used for explosive power, as also of the cylinder-cooling effect. In Col. 6 is given the approximate temperatures of explosion for a mixture of six air to one gas of 660 heat units per cubic foot, for the relative values of the clearance ratio in Col. 2 at constant volume.

Factors Governing Economy.—In view of the experiments in this direction, it clearly shows that in practical work, to obtain the greatest economy per effective brake-horsepower, it is necessary: 1st. To transform the heat into work with the greatest rapidity mechanically allowable. This means high piston speed. 2nd. To have high initial compression. 3rd. To reduce the duration of contact between the hot gases and the cylinder walls to the smallest amount possible; which means short stroke and quick speed, with a spherical cylinder head. 4th. To adjust the temperature of the jacket water (in water-cooled engines) to obtain the most economical output of actual power. This means heat radiating water-coils or honeycomb radiator structures with air-cooling surfaces suitable and adjustable to the most economical requirement of the engine, which by late trials requires the jacket water to be discharged at about 180° F. at sea level. In the case of air-cooled engines the radiating fins and the amount of surface exposed to the air stream providing the cooling effect must be carefully proportioned to prevent overcooling, rather than the impression that ordinarily obtains that such engines are apt to overheat. 5th. To reduce the wall surface of the clearance space or combustion chamber to the smallest possible area, in proportion to its required volume. This lessens the loss of the heat of combustion by exposure to a large surface, and allows of a higher mean wall temperature to facilitate the heat of compression.

Losses in Wall Cooling.—In an experimental investigation of the efficiency of a gas-engine under variable piston speeds made in France, it was found that the useful effect increases with the velocity of the piston—that is, with the rate of expansion of the burning gases with mixtures of uniform volumes; so that the variations of time of complete combustion at constant pressure, and the variations due to speed, in a way compensate in their efficiencies. The dilute mixture, being slow burning, will have its time and pressure quickened by increasing the speed.

Careful trials give unmistakable evidence that the useful effect increases with the velocity of the piston—that is, with the rate of expansion of the burning gases. The time necessary for the explosion to become complete and to attain its maximum pressure depends not only on the composition of the mixture, but also upon the rate of expansion. This has been verified in experiments with a high-speed motor, at speeds from 500 to 2,000 revolutions per minute, or piston speeds of from 16 to 64 feet per second. The increased speed of combustion due to increased piston speed is a matter of great importance to builders of gas-engines, as well as to the users, as indicating the mechanical direction of improvements to lessen the wearing

strain due to high speed and to lighten the reciprocating parts with increased strength, in order that the balancing of high-speed engines may be accomplished with the least weight.

From many experiments made in Europe and in the United States, it has been conclusively proved that excessive cylinder cooling by the water-jacket results in a marked loss of efficiency. In a series of early experiments with a Simplex engine in France, it was found that a saving of seven per cent in gas consumption per brake-horsepower was made by raising the temperature of the jacket water from 141° to 165° F. A still greater saving was made in a trial with an Otto engine by raising the temperature of the jacket water from 61° to 140° F.—it being 9.5 per cent less gas per brake-horsepower. This fact, which shows that high operating temperatures are desirable, probably accounts for the high efficiency of our modern air-cooled aircraft engine which can be run hotter than water-cooled engines without damaging the engine.

It has been stated that volumes of similar cylinders increase as the cube of their diameters, while the surface of their cold walls varies as the square of their diameters; so that for large cylinders the ratio of surface to volume is less than for small ones. This points to greater economy in the larger engines. The study of many experiments goes to prove that combustion takes place gradually in the gas-engine cylinder, and that the rate of increase of pressure or rapidity of firing is controlled by dilution and compression of the mixture, as well as by the rate of expansion or piston speed. The rate of combustion also depends on the size and shape of the explosion chamber, and is increased by the mechanical agitation of the mixture during compression and combustion, this being known as turbulence, and still more by the mode of firing, two or more sparks giving more power than a single sparkplug.

Distribution of Heat in a High-Speed Engine.—The following test results show the observed distribution of heat in the Ricardo variable-compression engine under various conditions of operation. Since they were taken in circumstances that make for a very high degree of accuracy, and under a fairly wide range of conditions, they are perhaps of some interest. With the exception of the tests on hydrogen, when the power was controlled by reducing the fuel supply and therefore the flame temperature, there is nothing novel. In all cases, except with hydrogen, the tests were made at a mixture strength ranging from five to ten per cent weak. The tests were carried out in three groups. Group A comprised tests at a constant compression-ratio and constant fuel-air ratio, but with varying speed, on two fuels; also, one series of tests at a higher compression ratio. Group B included tests at constant speed, but with the mean effective pressure varied by throttling, on two fuels. Group C was composed of tests at constant compression-ratio and constant speed, but with the mean effective pressure varied by varying the fuel-air ratio as in a Diesel engine; hydrogen gas being used as fuel in this instance. In all cases the following precautions were observed:

(1) The circulating water was maintained at a constant temperature of 60 degrees Centigrade, plus or minus two degrees Centigrade (140 degrees Fahrenheit, plus or minus 3.6 degrees Fahrenheit).

(2) The heat-imput to the carburetor was adjusted to bear a constant proportion to the weight of fuel consumed, except in the case of hydrogen when the carburetor was unheated.

(3) In Groups A and B, the fuel-air ratio in all cases was such as to give approximately ten per cent excess of air over that required for complete combustion of the fuel, the air consumption being measured and adjusted in each case.

(4) No readings were recorded until all temperature conditions had been stabilized for a considerable period after each change of state.

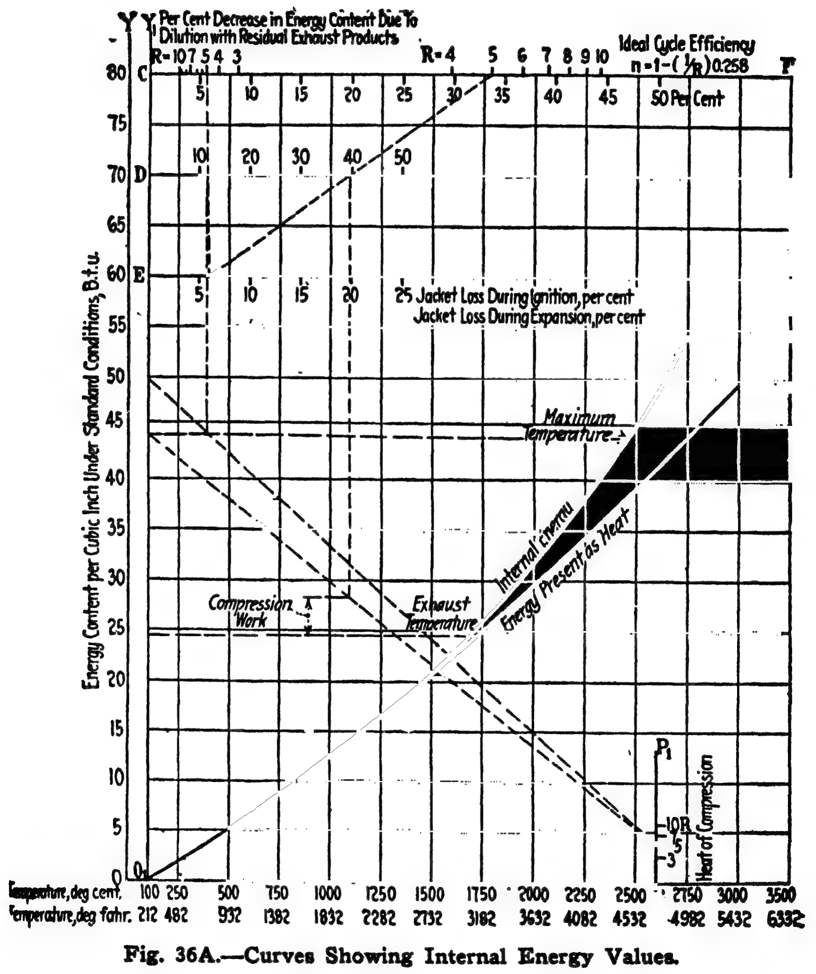


Fig. 36A.—Curves Showing Internal Energy Values.

The amount of heat dissipated, to the exhaust, by radiation and the like, is arrived at by difference in each case.

In all cases the indicated thermal efficiency can be taken as accurate to within about 0.5 per cent, and the heat to cooling water to within one per cent. Also, in all cases the heat produced by piston friction and that lost by radiation balanced at approximately 1,500 r.p.m. The water tem-

perature at which the readings were taken was that which the cylinder attains when motored continuously at 1,500 r.p.m.; that is, 45 or 60 degrees Centigrade (113 or 140 degrees Fahrenheit) above atmospheric temperature.

RICARDO TESTS SHOWING HEAT DISTRIBUTION IN A HIGH-SPEED INTERNAL-COMBUSTION ENGINE

Group A—Compression-Ratio, 3.8 to 1; Fuel, 95-Per Cent Ethyl Alcohol

Engine Speed, r.p.m.	975	1,300	1,500	1,700
Piston Speed, ft. per min.	1,300	1,733	2,000	2,266
Heat to Indicated Horsepower, per cent	26.9	27.0	26.9	27.0
Heat to Cooling Water, per cent	25.1	24.7	24.4	24.2
Heat to Exhaust, Radiation, etc., per cent	48.0	48.3	48.7	48.8
Total Heat, per cent	100.0	100.0	100.0	100.0

Group A—Compression-Ratio, 3.8 to 1; Fuel, Grade A Gasoline

Engine Speed, r.p.m.	975	....	1,500	1,200
Piston Speed, ft. per min.	1,300	....	2,000	2,266
Heat to Indicated Horsepower, per cent	25.9	....	26.1	26.1
Heat to Cooling Water, per cent	30.4	....	28.0	27.0
Heat to Exhaust, Radiation, etc., per cent	43.7	....	45.9	46.9
Total Heat, per cent	100.0	....	100.0	100.0

Group A—Compression-Ratio, 7 to 1; Fuel, 95-Per Cent Ethyl Alcohol

Engine Speed, r.p.m.	975	1,300	1,500	....
Piston Speed, ft. per min.	1,300	1,733	2,000	....
Heat to Indicated Horsepower, per cent	37.6	38.1	38.3	....
Heat to Exhaust, Radiation, etc., per cent	37.0	37.6	37.8	....
Heat to Cooling Water, per cent	25.4	24.3	23.9	....
Total Heat, per cent	100.0	100.0	100.0	....

Group B—Throttling at Constant Fuel-Air Ratio. Compression-Ratio, 5.45 to 1; Engine Speed, 1,500 r.p.m.; Piston Speed, 2,000 ft. per min.; Fuel, 95-Per cent Ethyl Alcohol

Percentage of Maximum Indicated Horsepower	100.0	80.0	60.0	40.0
Heat to Indicated Horsepower, per cent	34.8	35.0	35.0	34.8
Heat to Cooling Water, per cent	24.1	26.0	29.2	33.0
Heat to Exhaust, Radiation, etc., per cent	41.1	39.0	35.8	32.2
Total Heat, per cent	100.0	100.0	100.0	100.0

Group B—Compression-Ratio, 5.45 to 1; Engine Speed, 1,500 r.p.m.; Piston Speed, 2,000 ft. per min.; Fuel, Grade A Gasoline

Percentage of Maximum Indicated Horsepower	100.0	80.0	60.0	40.0
Heat to Indicated Horsepower, per cent	33.5	34.0	34.1	33.5
Heat to Cooling Water, per cent	26.5	28.2	31.8	35.5
Heat to Exhaust, Radiation, etc., per cent	40.0	37.8	34.1	31.0
Total Heat, per cent	100.0	100.0	100.0	100.0

Group C—Mean Effective Pressure Varied by Varying Fuel-Air Ratio from 15 Per Cent Excess of Air Upward; Compression-Ratio, 5.45 to 1; Engine Speed, 1,500 r.p.m.; Piston Speed, 2,000 ft. per min.; Fuel, Hydrogen

Percentage of Maximum Indicated Horsepower	100.0	80.0	60.0	40.0
Heat to Indicated Horsepower, per cent	33.3	35.6	38.2	40.0
Heat to Cooling Water, per cent	23.6	24.9	25.3	28.6
Heat to Exhaust, Radiation, etc., per cent	43.1	39.5	36.5	31.4
Total Heat, per cent	100.0	100.0	100.0	100.0

Internal Energy Values.—In Fig. 36 A the internal energy curve is plotted in terms of British thermal units per standard cubic inch on a vertical scale, against the temperature on a horizontal scale by H. R. Ricardo. The other full-line curve shows the energy present as heat, so that the difference between the two curves shows the chemical energy stored in the products of dissociation. Zero energy is taken at 100 degrees Centigrade (212 degrees Fahrenheit) as being an average temperature at the beginning of compression. Variations in this temperature will have but little influence. The explanation given below of the use of the diagram is supplemented by an example worked out for the following data, the construction lines of the example being shown dotted in Fig. 36 A.

Compression-ratio, R,	= 5 to 1
Energy content	= 46.2 B.t.u. per cubic inch
Heat loss during combustion	= 6 per cent
Heat loss during expansion	= 6 per cent

There are three factors in an actual engine which modify the temperature attained by the combustion of a mixture of any given energy content. They are the

(1) Heat put into the mixture by compression
(2) Loss due to cooling by the walls of the combustion-chamber during combustion
(3) Effective weakening of the mixture due to dilution with the residual exhaust products

Factor (1) is allowed for by laying off the heats of compression for various ratios by the marks "R = 5," etc., on the line P P_1 near the bottom of the diagram. The energy content is then marked off above this on the vertical line O_1 Y_1 representing the 100 degrees Centigrade (212 degrees Fahrenheit) starting-point. In the example, the 46.2 B.t.u. energy content is laid off above the 3.6 B.t.u. of compression, making a total of 49.8 B.t.u., this being the gross energy content from which the losses due to factors (2) and (3) must be deducted. This is done in the following manner:

On the horizontal scale C is marked the effective energy loss due to dilution with residual exhaust, assumed to be at 1,000 degrees Centigrade (1,832 degrees Fahrenheit). Scale E shows the percentage loss due to cooling during combustion. This is laid off at any figure which previous experience shows as probable for the type of combustion-chamber in question; this is six per cent in the example. A line is then drawn between these two points, and the point of intersection of this line with the scale D gives the total percentage loss due to these two causes; this is 11.5 per cent in the example.

To transfer this to the diagram, a line is dropped vertically from the above intersection point. Another line is drawn from the point on the line O_1 Y_1 giving the gross British thermal units per cubic inch to the suitable compression point on the line P P_1, and representing 100 per cent on scale D; this is 49.8 B.t.u. per cubic inch in the example. From the intersection of the above two lines, a horizontal line is run to the energy scale on one side, and the energy curve on the other. The point on the energy scale shows the net energy available for expansion; it is 44.5 B.t.u. per cubic inch

in the example. From the energy-temperature curve, the actual flame temperature can be read off; this is 2,475 degrees Centigrade (4,487 degrees Fahrenheit) in the example.

The drop in temperature during the expansion-stroke depends on the two factors of (a), external work done and (b) heat loss to the walls. The net power output is given as a percentage of the heat content of the mixture on scale F, the formula used being $n = 1 - (1/R)^{0.258}$. This covers all dissociation and similar effects, but not wall losses. The wall loss during expansion is laid off on scale E. A line drawn between these point gives their sum on scale D as before. A perpendicular from this point is dropped to meet a line from the net-energy point on O_1 Y_1 to the suitable-compression point on the line P P_1. As the gross work done during expansion is the sum of the net work mentioned above and the compression work, this latter amount, which is 3.6 B.t.u. in the example, must be laid off below the above intersection point, to find the energy content at the end of expansion, which is 24.5 B.t.u. in the example. The corresponding final temperature, 1,675 degrees Centigrade (3,047 degrees Fahrenheit) in the example, can then be read off from the energy curve.

Taking an actual example from the variable-compression-engine data, with a correct mixture of an energy content of 46.2 B.t.u. per cubic inch and a compression-ratio of five to one, the actual maximum flame-temperature, as obtained from the diagram in Fig. 35 C allowing for the additional heat of compression, the wall loss during combustion and the dilution by residual exhaust products, will be 2,475 degrees Centigrade (4,487 degrees Fahrenheit), corresponding to an energy content of 44.5 B.t.u. per standard cubic inch. At a ratio of five to one, the observed indicated thermal efficiency is 31 per cent; of this, five per cent is due to the change in specific volume of the mixture, so that the heat drop is $46.2 \times 31 \times 100/105 = 13.6$ B.t.u. per cubic inch. Add to this the 3.6 B.t.u. of compression work restored during expansion, and the six per cent of 46.2 or 2.8 B.t.u. of wall loss during expansion, and the total heat-drop during expansion becomes 20.0 B.t.u. per cubic inch leaving a final energy content of 24.5 B.t.u. per cubic inch which, it will be observed, coincides with the figure found in the example under the same conditions. The corresponding final temperature is 1,675 degrees Centigrade (3,047 degrees Fahrenheit).

While affecting the final temperature directly, it should be observed that the loss of heat during expansion has only a slight influence on the actual efficiency; this has been ignored in the construction, because much of it is lost late in the expansion-stroke where its value is less. Another slight error allowed to remain in the construction, for the sake of simplicity, is that a percentage of the net heat available during combustion is deducted for the jacket loss during expansion; whereas this is given as a proportion of the total heat available in the fuel. The error due to this cause is however very small, being in the case considered $2.8\ (46.2 - 44.5) \div 44.5 = 0.11$ B.t.u. per cubic inch, and can be ignored safely.

Improving Engine Performance.—Results of three sets of investigations looking to the improvement of internal-combustion-engine performance are set forth in a paper by H. M. Jacklin of Purdue University, West Lafayette, Ind., published in the March, 1928 *S. A. E. Journal,* entitled "Im-

proving Engine Performance." The first set was with multiple ignition, the second with high compression, and the third with the comparative behavior of a variable-compression engine when operating as a constant-clearance engine and when operating with constant compression. The tests with multiple ignition indicate that it increases the power about nine to ten per cent at full throttle and generally gives smoother operation than ignition with a single sparkplug, especially on the leaner air-fuel mixtures. Increasing the compression ratio from 5.3 to 1 to 10 to 1 resulted in a thirteen per cent increase in the power developed by the engine.

Operating the engine on the constant-compression principle resulted in a fuel saving of as much as 34 per cent. The thermal efficiency was in-

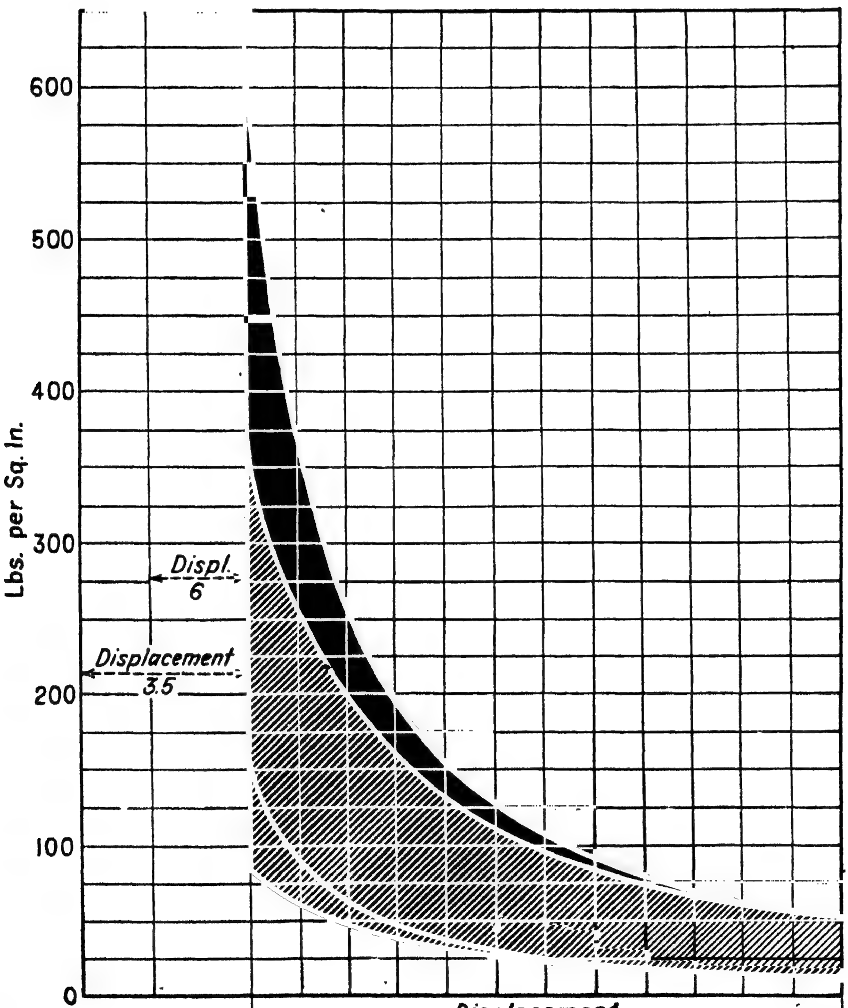


Fig. 37.—Theoretical Cylinder Diagrams for Compression Ratio of 4, 5 and 7 Showing a Big Increase in Pressure with Augmented Compression.

creased in the same amount, and the exhaust temperatures dropped more rapidly at reduced loads under constant compression than in the conventional constant-clearance operation. Fixed spark is entirely feasible under constant-compression operation. Combustion is much better than it is with constant-clearance operation and this should result in less carbon deposition and less crankcase-oil dilution.

Writing on this subject in *Automotive Industries*, P. M. Heldt, the well known automotive engineering authority says that automotive engines can be made more efficient by increasing their ratio of expansion. In the ordinary engine the expansion ratio is equal to the compression ratio, and the theoretical thermal efficiency then depends upon the compression ratio. We can increase the expansion ratio in two ways—by increasing the compression ratio, and by expanding the burning charge to a volume larger than that occupied by it before compression. With present day commercial fuels it is not practical to increase the compression ratio beyond 4.5 to 1, or at most 5 to 1, as a higher rate will result in detonation. But if anti-detonating fuels should find a wide market and their distribution become general, it is not inconceivable that automotive engine builders would design their engines to operate at higher compressions, to take full advantage of the qualities of the fuel. With fuels of the characteristics of ethyl gasoline, for instance, a compression ratio of seven to one would be quite practicable. Certain blends of aviation gasoline permit compression ratios of six to one and a number of the newer engines use that ratio.

Effect of Increasing Compression Ratio.—In Fig. 37 are shown theoretical indicator diagrams for cylinders with compression ratios of 4.5 and 7 respectively. The corners have not been taken off, as they would be in an actual diagram, because the effect on the areas of the two diagrams would be the same, and only comparative results are aimed at here. The energy in ft.-lb. represented by each diagram is given by the equation

$$W = \frac{D\,P_1}{3.6}\,(a - 1)\,\frac{r^{1.3} - r}{r - 1},$$

where D is the piston displacement in cubic inches; a the ratio of pressure multiplication on ignition; P_1 the initial pressure in the cylinder at the beginning of the compression stroke, and r the compression ratio. Placing P_1 at 12 lb. p. sq. in. absolute and a at 4.25, we get for the energy developed per charge with a compression ratio of 4.5

$$W = \frac{D \times 12}{3.6} \times (4.25 - 1) \times \frac{7.066 - 4.5}{4.5 - 1} = 10.82\,D \times 0.733 = 7.93\,D \text{ ft.-lb.}$$

Similarly, for a compression ratio of 7 we get

$$W = \frac{D \times 12}{3.6} (4.25 - 1) \times \frac{12.55 - 7}{7 - 1} = 10.82\,D \times 0.925 = 10.00\,D \text{ ft.-lb.}$$

These figures indicate a gain of 26 per cent in energy per charge and hence, since the amount of fuel taken in per charge is the same, an increase in the thermal efficiency of 26 per cent. That the figure arrived at by this method

is not the same as that arrived at by the former method is evidently due to the fact that the equation for the air-cycle efficiency does not take account of the energy consumed in compression.

The above figures given by Mr. Heldt apply directly to the gain in indicated energy, and the relation of the indicated horsepowers would be the same. Although the gas pressures are somewhat higher with the higher compression, there is no reason for expecting a material increase in the friction losses. In fact, it is quite reasonable to assume that the friction losses will be substantially the same in both cases. To obtain the brake-horsepowers we would therefore have to subtract the same figure from the two indicated horsepowers and it is obvious that the proportional gain in brake-horsepower would be greater than the gain in indicated horsepower. It would therefore appear that an increase in brake-horsepower of from 20 to 25 per cent would result from an increase in the compression ratio from 4.5 to 7, and since the fuel consumption would remain the same, the brake thermal efficiency would increase in the same proportion.

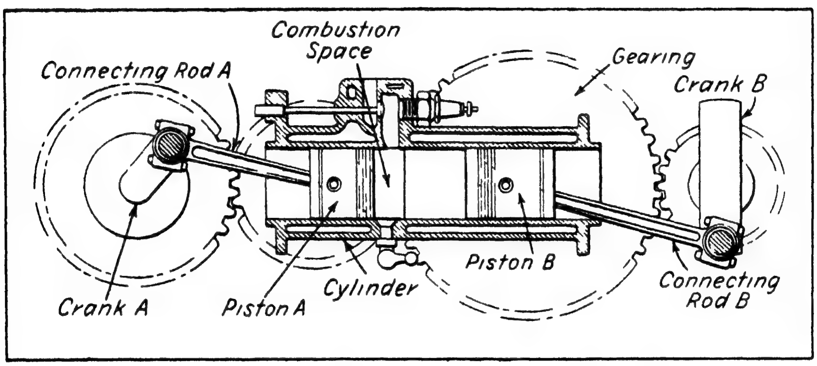


Fig. 38.—Diagram Showing Action of Double Piston Engine with Inlet and Expansion Strokes Longer Than the Exhaust and Compression Strokes.

Use of Long Expansion Stroke.—The other method of increasing the expansion ratio described by Mr. Heldt is by continuing the expansion beyond the volume which the charge occupied when at atmospheric pressure previous to compression. In the early days of the gas-engine, Atkinson invented a compound crank mechanism whereby the piston would perform alternately two long and two short strokes, the expansion and exhaust strokes being long and the inlet and compression strokes short. The idea was, of course, to expand the gases more nearly to atmospheric pressure and thus utilize the power remaining in them in the ordinary engine when the exhaust valve is lifted and they are allowed to escape to the atmosphere.

This idea of increasing the expansion by using unequal inlet and expansion strokes has been revived at intervals, and quite recently the design shown in Fig. 38 was patented in England. Use is made of two pistons in each cylinder, each piston being connected to its own crankshaft. The two crankshafts are connected by gearing in such a manner that the upper

one or B crankshaft rotates at half the speed of the lower one designated as crankshaft A. During the inlet stroke the two pistons move in the same direction, but, one crank turning twice as fast as the other, the lower piston moves much more rapidly than the upper one and the distance between them increases though both are moving in the same direction. During the following compression stroke the two pistons move again in the same direction (in the opposite direction as during the inlet stroke), and thus the relative motion between them is small. Then follows the power stroke, during which the slow upper piston continues to move upward while the faster lower piston now moves downward. At the end of the power stroke both pistons are in their extreme positions, while at the beginning of this stroke the lower piston is at the top end of its stroke while the upper piston is at midstroke. Hence the effective stroke or expansion movement is equal to the full stroke of the lower or piston A plus one-half the stroke of the upper piston B.

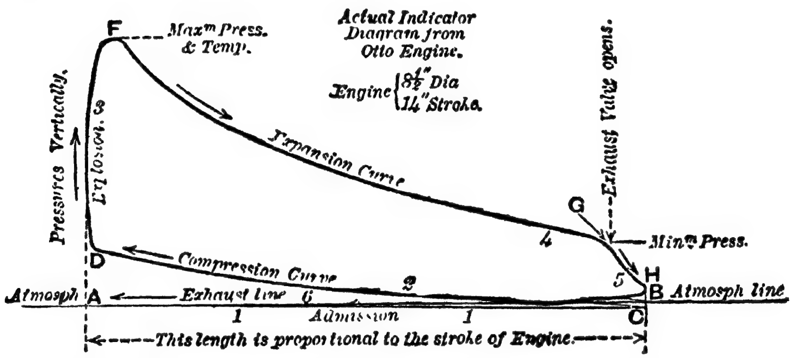


Fig. 39.—Indicator Card of Four-Cycle Engine Showing Pressure Variation at Various Points of the Otto Cycle.

The reader will realize, by studying the diagram at Fig. 38 that the type of engine depicted there would be undesirable for aircraft purposes because it has greater mechanical complication than is found in the simple engines. Its weight-horsepower ratio would not be favorable and even though there was a substantial gain in efficiency, it would not balance the added weight and complication of the design that permitted greater expansion of the exploded gas on the explosion stroke. It is also doubtful if such an engine could be run at the high operating speeds required in aircraft work so the design would seem limited to stationary applications rather than automotive uses. The use of twice as many reciprocating parts would call for niceties of balance that greatly reduce the possibilities of commercially utilizing such designs.

Value of Indicator Cards.—To the uninitiated, indicator cards are considerable of a mystery; to those capable of reading them they form an index relative to the action of any engine. An indicator card, such as shown at Fig. 39 is merely a graphical representation of the various pressures existing in the cylinder for different positions of the piston. The length is to some scale that represents the stroke of the piston. During the intake

stroke, the pressure falls below the atmospheric line. During compression, the curve gradually becomes higher owing to increasing pressure as the volume is reduced. After ignition the pressure line moves upward almost straight, then as the piston goes down on the explosion stroke, the pressure falls gradually to the point of exhaust valve opening, when the sudden release of the imprisoned gas causes a reduction in pressure to nearly atmospheric. An indicator card, or a series of them, will always show by its lines the normal or defective condition of the inlet valve and passages; the actual line of compression; the firing moment; the pressure of explosion; the velocity of combustion; the normal or defective line of expansion, as measured by the adiabatic curve, and the normal or defective operation of the exhaust valve, exhaust passages, and exhaust pipe. In fact, all the cycles of an explosive motor may be made a practical study from a close investigation of the lines of an indicator card.

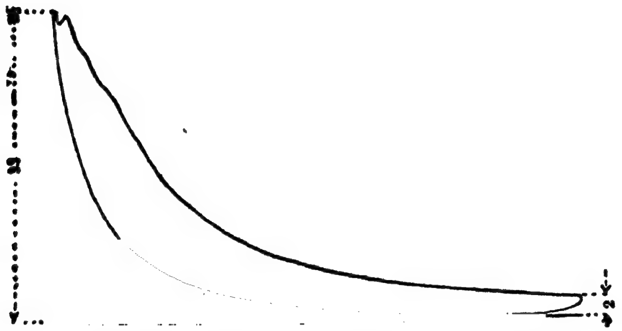


Fig. 40.—Indicator Card of Diesel Motor.

A most unique card is that of the Diesel motor (Fig. 40), which involves a distinct principle in the design and operation of internal-combustion motors, in that instead of taking a mixed or carbureted charge for instantaneous explosion, its charge primarily is of air and its compression to a pressure at which a temperature is attained above the igniting point of the fuel, then injecting the fuel under a still higher pressure by which spontaneous combustion takes place gradually with increasing volume over the compression for part of the stroke or until the fuel charge is consumed. The motor thus operating between the pressures of 500 and 35 lbs. per square inch, with a clearance of about seven per cent, has given an efficiency of 36 per cent of the total heat value of fuel oil. The action of such engines in their various forms has been fully considered in preceding chapters.

Value of Compression in Explosive Motors.—That the compression in a gas-, gasoline-, or oil-engine has a direct relation to the power obtained, has been long known to experienced engine designers and builders, having been suggested by M. Beau de Rocha, in 1862, and afterward brought into practical use in the four-cycle or Otto type as early as 1880 which indicates that some of the operating factors of our modern automotive engines were realized practically fifty years ago. This is twice the life span of the airplane that the internal-combustion engine made possible. The degree

of compression has had a growth from zero, in the early engines, to the highest available due to the varying ignition temperatures of the different gases and vapors used for explosive fuel, in order to avoid premature explosion from the heat of compression. Much of the increased power for equal-cylinder capacity is due to compression of the charge from the fact that the most powerful explosion of gases, or of any form of explosive material, takes place when the particles are in the closest contact or cohesion with one another, less energy in this form being consumed by the ingredients themselves to bring about their chemical combination, and consequently more energy is given out in useful or available work. This is best shown by the ignition of gunpowder, which, when ignited in the open air, burns rapidly, but without explosion, an explosion only taking place if the powder be confined or compressed into a small space.

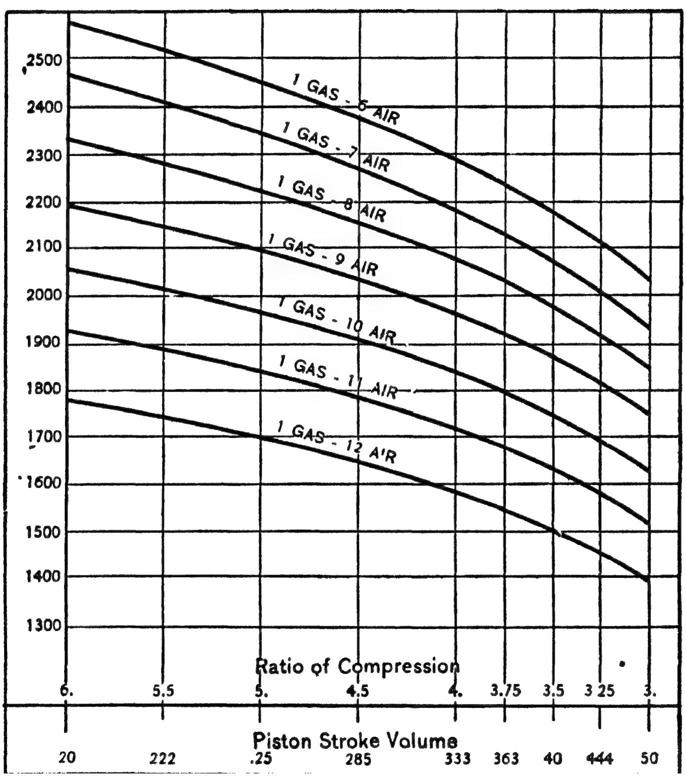


Fig. 41.—Diagrams Showing Heat in Gas-Engine Cylinder Obtained by the Combustion of Various Gas and Air Mixtures.

In a gas- or gasoline-motor with a small clearance or compression space —with high compression—the surface with which the burning gases come into contact is much smaller in comparison with the compression space in a low-compression motor. Another advantage of a high-compression motor is that on account of the smaller clearance of combustion space less cooling water is required than with a low-compression motor, as the temperature,

and consequently the pressure, falls more rapidly. The loss of heat through the water-jacket is thus less in the case of a high-compression than in that of a low-compression motor. In the noncompression type of motor the best results were obtained with a charge of sixteen to eighteen parts of gas and 100 parts of air, while in the compression type the best results are obtained with an explosive mixture of seven to ten parts of gas and 100 parts of air, thus showing that by the utilization of compression a weaker charge with a greater thermal efficiency is permissible.

It has been found that the explosive pressure resulting from the ignition of the charge of gas or gasoline-vapor and air in the gas-engine cylinder is about four and one-half times the pressure prior to ignition. The difficulty about getting high compression is that if the pressure is too high the charge is likely to ignite prematurely, as compression always results in increased temperature. The cylinder may become too hot, a deposit of carbon, a projecting electrode or plug body in the cylinder may become incandescent and ignite the charge which has been excessively heated by the high compression and mixture of the hot gases of the previous explosion.

Factors Limiting Compression.—With ordinary gasoline-vapor and air the compression should not be raised above about 90 to 95 pounds to the square inch, many manufacturers not going above 70 or 80 pounds, though this figure may be greatly augmented by using ethyl-gasoline or even benzol-gasoline blends which produce anti-knock fuels, as previously outlined, the chart at Fig. 37 showing that pressures of 150 pounds are possible prior to ignition. For natural gas in stationary engines the compression pressure may easily be raised to from 85 to 100 pounds per square inch. For gases of low calorific value, such as blast-furnace or producer-gas, the compression may be increased to from 140 to 190 pounds. In fact the ability to raise the compression to a high point with these gases is one of the principal reasons for their successful adoption for large gas-engine use. In kerosene injection engines the compression of 250 pounds per square inch has been used with marked economy. Many troubles in regard to loss of power and increase of fuel have occurred and will no doubt continue, owing to the wear of valves, piston, and cylinder, which produces a loss in compression and explosive pressure and a waste of fuel by leakage. Faulty adjustment of valve movement is also a cause of loss of power; which may be from tardy closing of the inlet valve or a too early opening of the exhaust valve. The explosive pressure in all forms of internal-combustion engines varies to a considerable amount in proportion to the compression pressure by the difference in fuel value and the proportions of air mixtures, so that for good illuminating gas the explosive pressure may be from 2.5 to 4 times the compression pressure. For natural gas 3 to 4.5, for gasoline 3 to 5, for producer-gas 2 to 3, for kerosene by injection 3 to 6, and for ethyl-gasoline from 4 to 6.

The compression temperatures, although well known and easily computed from a known normal temperature of the explosive mixture, are subject to the effect of the uncertain temperature of the gases of the previous explosion remaining in the cylinder, the temperature of its walls, and the relative volume of the charge, whether full or scant; which are terms too variable to make any computations reliable or available. For the theoreti-

cal compression temperatures from a known normal temperature, we append a study of the rise in temperature for the compression pressures in the following table:

TABLE IV

Compression Temperatures from a Normal Temperature of '60 Degrees Fahrenheit

100 lbs. gauge	484°	60 lbs. gauge	373°
90 lbs. gauge	459°	50 lbs. gauge	339°
80 lbs. gauge	433°	40 lbs. gauge	301°
70 lbs. gauge	404°	30 lbs. gauge	258°

Chart for Determining Compression Pressures.—A very useful chart (Fig. 42) for determining compression pressures in gasoline-engine cylinders for various ratios of compression space to total cylinder volume is given by P. S. Tice, and described in the Chilton Automobile Directory by the originator as follows:

"It is many times desirable to have at hand a convenient means for at once determining with accuracy what the compression pressure will be in a gasoline-engine cylinder, the relationship between the volume of the compression space and the total cylinder volume or that swept by the piston being known. The curve at Fig. 42 is offered as such a means. It is based on empirical data gathered from upward of two dozen modern automobile engines and represents what may be taken to be the results as found in practice. It is usual for the designer to find compression pressure values, knowing the volumes from the equation

$$P_2 = P_1 \left(\frac{V_1}{V_2} \right)^{1.4} \qquad (1)$$

which is for adiabatic compression of air. Equation (1) is right enough in general form but gives results which are entirely too high, as almost all designers know from experience. The trouble lies in the interchange of heat between the compressed gases and the cylinder walls, in the diminution of the exponent (1.4 in the above) due to the lesser ratio of specific heat of gasoline vapor and in the transfer of heat from the gases which are being compressed to whatever fuel may enter the cylinder in an unvaporized condition. Also, there is always some piston leakage, and, if the form of the equation (1) is to be retained, this also tends to lower the value of the exponent. From experience with many engines, it appears that compression reaches its highest value in the cylinder for but a short range of motor speeds, usually during the mid-range. Also, it appears that, at those speeds at which compression shows its highest values, the initial pressure at the start of the compression stroke is from .5 to .9 lb. below atmospheric. Taking this latter loss value, which shows more often than those of lesser value, the compression is seen to start from an initial pressure of 13.9 lbs. per sq. in. absolute.

"Also, experiment shows that if the exponent be given the value 1.26, instead of 1.4, the equation will embrace all heat losses in the compressed

gas, and compensate for the changed ratio of specific heats for the mixture and also for all piston leakage, in the average engine with rings in good condition and tight. In the light of the foregoing, and in view of results obtained from its use, the above curve is offered—values of P_2 being found from the equation

$$P_2 = 13.8\left(\frac{V_1}{V_2}\right)^{1.26}$$

"In using this curve it must be remembered that pressures are absolute. Thus: suppose it is desired to know the volumetric relationships of the cylinder for a compression pressure of 75 lbs. gauge. Add atmospheric pressure to the desired gauge pressure $14.7 + 75 = 89.7$ lbs. absolute. Locate this pressure on the scale of ordinates and follow horizontally across

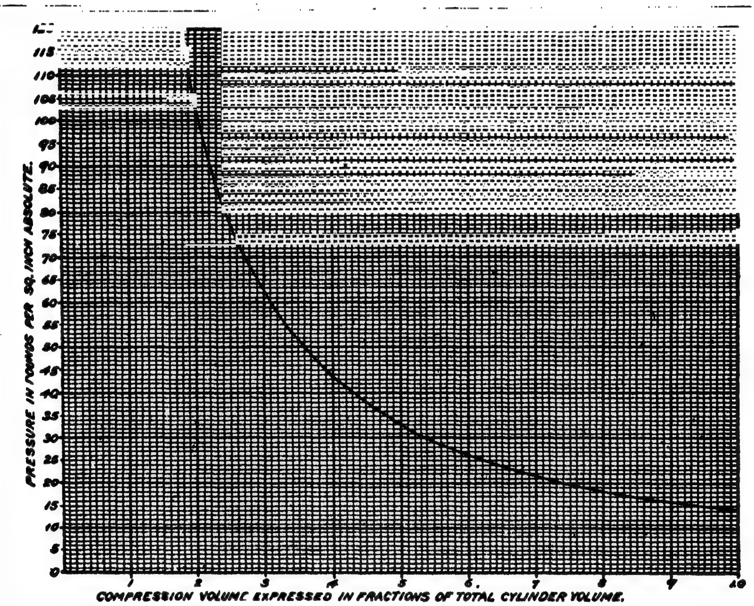


Fig. 42.—Chart Showing Relation Between Compression Volume Explained in Fraction of Total Cylinder Volume and Pressure in Pounds per Square Inch Absolute.

to the curve and then vertically downward to the scale of abscissas, where the ratio of the combustion chamber volume to the total cylinder volume is given, which latter is equal to the sum of the combustion chamber volume and that of the piston sweep. In the above case it is found that the combustion space for a compression pressure of 75 lbs. gauge will be .225 of the total cylinder volume, or $.225 \div 775 = .2905$ of the piston sweep volume. Conversely, knowing the volumetric ratios, compression pressure can be read directly by proceeding from the scale of abscissas vertically to the curve and thence horizontally to the scale of ordinates."

Causes of Heat Loss in Motors.—The difference realized in the practical operation of an internal-combustion heat engine from the computed effect derived from the values of the explosive elements is probably the most serious difficulty that engineers have encountered in their endeavors to arrive at a rational conclusion as to where the losses were located, and the ways and means of design that would eliminate the causes of loss and

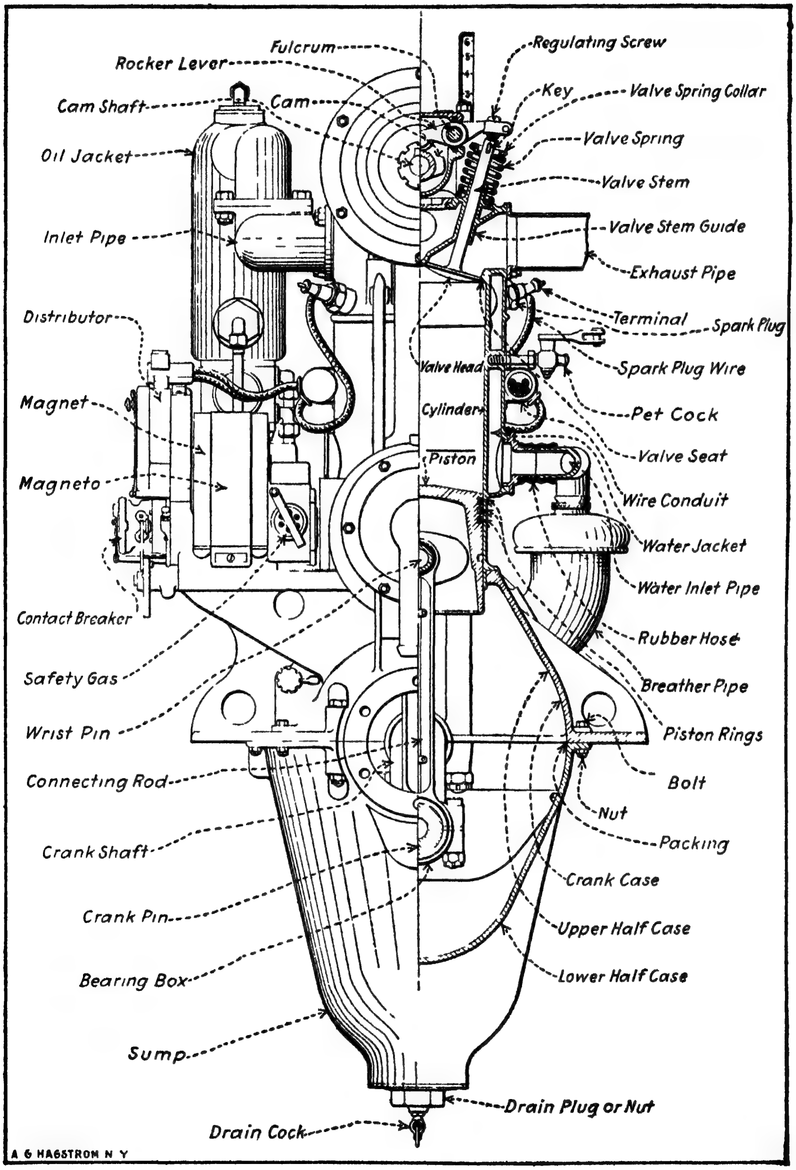
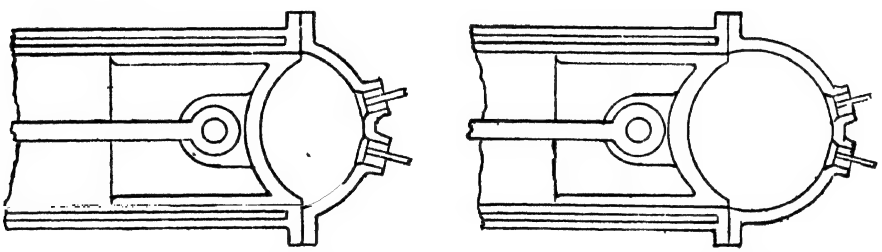


Fig. 43.—Part Sectional View of an Early Airplane Motor of Hall-Scott Design Showing Principal Parts and Combustion-Chamber Form.

raise the efficiency step by step to a reasonable percentage of the total efficiency of a perfect cycle. An authority on the relative condition of the chemical elements under combustion in closed cylinders attributes the variation of temperature shown in the fall of the expansion curve, and the suppression or retarded evolution of heat, entirely to the cooling action of the cylinder walls, and to this nearly all the phenomena hitherto obscure in the cylinder of a gas-engine. Others attribute the great difference between the theoretical temperature of combustion and the actual temperature realized in the practical operation of the gas-engine, a loss of more than one-half of the total heat energy of the combustibles, partly to the dissociation of the elements of combustion at extremely high temperatures and their reassociation by expansion in the cylinder, to account for the supposed continued combustion and extra adiabatic curve of the expansion line on the indicator card.



Figs. 44 and 45.—Diagram Showing Spherical Combustion-Chamber at Left and Greatly Enlarged Combustion-Chamber at Right.

Combustion Chamber Form Important.—The loss of heat to the walls of the cylinder, piston, and clearance space, as regards the proportion of wall surface to the volume, has gradually brought this point to its smallest ratio in the concave piston-head and globular cylinder-head, with the smallest possible space in the inlet and exhaust passage. The wall surface of a cylindrical clearance space or combustion chamber of one-half its unit diameter in length is equal to 3.1416 square units, its volume but 0.3927 of a cubic unit; while the same wall surface in a spherical form has a volume of 0.5236 of a cubic unit. It will be readily seen that the volume is increased 33⅓ per cent in a spherical over a cylindrical form for equal wall surfaces at the moment of explosion, when it is desirable that the greatest amount of heat is generated, and carrying with it the greatest possible pressure from which the expansion takes place by the movement of the piston.

The spherical form cannot continue during the stroke for mechanical reasons; therefore some proportion of piston stroke of cylinder volume must be found to correspond with a spherical form of the combustion chamber to produce the least loss of heat through the walls during the combustion and expansion part of the stroke. This idea is illustrated in Figs. 44 and 45, showing how the relative volumes of cylinder stroke and combustion chamber may be varied to suit the requirements due to the quality of the elements of combustion.

Although the concave piston-head shows economy in regard to the relation of the clearance volume to the wall area at the moment of explosive

combustion, it may be clearly seen that its concavity increases its surface area and its capacity for absorbing heat, for which there is no provision for cooling the piston, save its contact with the walls of the cylinder and the slight air cooling of its back by its reciprocal motion. For this reason the concave piston-head has not been generally adopted and the concave cylinder-head, as shown in Fig. 45, with a flat piston-head is the latest and best practice in airplane engine construction. The practical application of

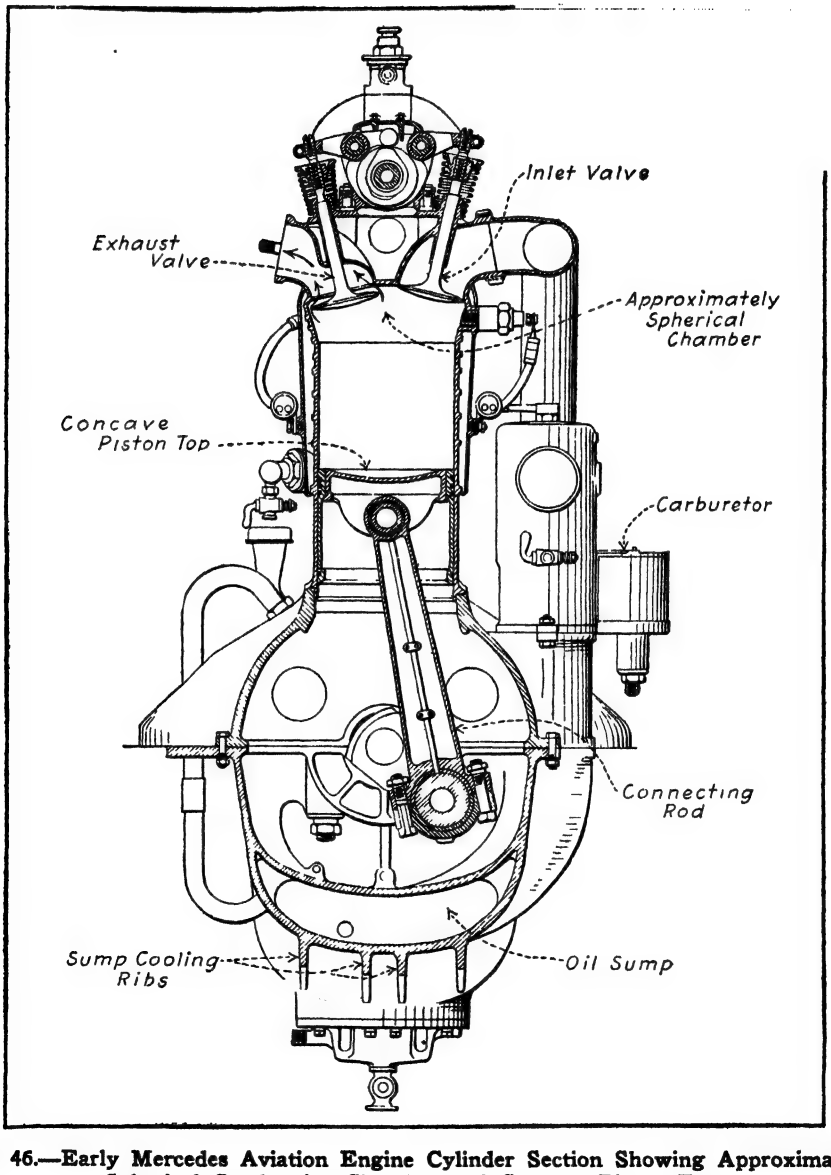


Fig. 46.—Early Mercedes Aviation Engine Cylinder Section Showing Approximately Spherical Combustion-Chamber and Concave Piston Top.

the principle just outlined to one of the most efficient water-cooled airplane motors ever designed, the Mercedes, is clearly outlined at Fig. 46. It will be evident that practical considerations of valve size and location make a really spherical head difficult to realize in practice and rather favor the roof head as will be considered later.

Heat Losses to Cooling Water.—The mean temperature of the wall surface of the combustion chamber and cylinder, as indicated by the temperatures of the circulating water, has been found to be an important item in the economy of the gas-engine. Dugald Clerk, in England, an early and highly regarded authority in practical work with the gas-engine, found that ten per cent of the gas for a stated amount of power was saved by using water at a temperature in which the ejected water from the cylinder-jacket was near the boiling point, and ventures the opinion that a still higher temperature for the circulating water may be used as a source of economy. This principle is made use of by present day advocates of steam cooling and also is quoted by proponents of air-cooled engines. This could be made practical in the case of aviation engines by adjusting the air-cooling surface of the radiator so as to maintain the inlet water at just below the boiling point, and by the rapid circulation induced by the pump pressure, to return the water from the cylinder-jacket a few degrees above the boiling point. The thermal displacement systems of cooling employed in automobile engines result in their working under more favorable temperature conditions than those engines in which cooling is more energetic. Care must be taken, however, to keep the maximum temperature of the jacket water enough below boiling point at sea level so it will not boil away at altitudes where the boiling point becomes lower than at sea level. Water outlet temperature is usually 180 to 190 degrees Fahrenheit at sea level.

For a given amount of heat taken from the cylinder by the largest volume of circulating water, the difference in temperature between inlet and outlet of the water-jacket should be the least possible, and this condition of the water circulation gives a more even temperature to all parts of the cylinder; while, on the contrary, a cold-water supply, say at 60° F., so slow as to allow the ejected water to flow off at a temperature near the boiling point, must make a great difference in temperature between the bottom and top of the cylinder, with a loss in economy in gas and other fuels, as well as in water, if it is obtained by measurement.

From the foregoing considerations of losses and inefficiencies, we find that the practice in aircraft motor design and construction has not yet reached the desired perfection in its cycular operation. Step by step improvements have been made with many changes in design though many have been without merit as an improvement, farther than to gratify the longings of designers for something different from the other thing, and to establish a special construction of their own. These efforts may in time produce a motor of normal or standard design for each kind of fuel that will give the highest possible efficiency for all conditions of service.

Horsepower Increase by Higher Speeds.—In automobile racing applications, increasing the speed has been a comparatively easy way of securing more power out of a given displacement, the horsepower being theoretically proportional to the speed (that is, the number of revolutions). Of course

a certain amount of ingenuity has had to be used and considerable work done to get engines to run at the present high speeds. Col. W. G. Wall has stated that in 1912 the Indianapolis 500-mile race was won by a car with an engine of 490 cu. in. displacement, which turned over at a maximum speed of about 2,200 r.p.m. The fast cars in the 1927 race had a piston displacement of 91 cu. in., less than one-fifth the size of the larger one, with a maximum engine speed of about 7,500 r.p.m. The former engine developed about 100 hp. maximum, or about 0.204 hp. per cu. in. displacement, while the latter developed 160 hp. or more, or about 1.75 hp. per cu. in. displacement.

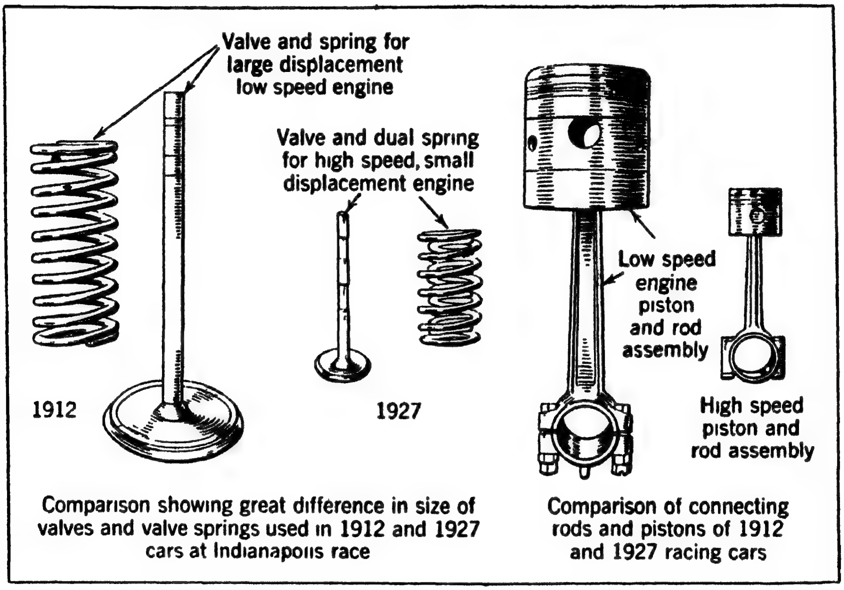


Fig. 47.—Diagram Showing Comparison Between Size of Parts of Low-Speed Large Displacement Engines Compared to Equivalent Parts of High-Speed Small Displacement Engines. Showing How Weight per Horsepower can be Reduced by Increasing Crankshaft Revolutions and Diminishing Cylinder Contents.

Less than fifteen years ago 3,000 r.p.m. for an engine was considered extremely high speed and many engineers thought even that speed was too fast for an engine to withstand. Today there are a number of passenger car engines that will turn up to 4,500 r.p.m., and racing car engines do up to 8,000 r.p.m. Colonel Wall stated that a certain foreign manufacturer has developed a reciprocating engine which turns over at 11,000 r.p.m. It thus seems that the limit of speed may be very much higher than we today can well imagine. There is a practical limit, however, to all engine speed and it has been necessary, in order to make the reciprocating parts sufficiently light, to make the bore of the cylinders smaller and the pistons lighter. Alloy steels and aluminum alloys have played an important part in this work, especially aluminum pistons and in some cases forged duralumin connecting rods.

It has been necessary to develop high pressure oiling systems, to lubri-

cate not only the bearings which are revolving so fast, but also the cylinders, so as to cut down the great amount of friction there would otherwise be as we must remember that the pistons are traveling through the bore of the cylinders in these high-speed engines at the rate of about a mile a minute.

Supercharger Makes High Speed Possible.—The high-speed engine of today is due largely to the supercharger. In order to run up speed, it is absolutely necessary to cut down the weight of the reciprocating parts. One of the easiest ways of doing this is to cut down the bore of the cylinders, but this also means that the size of the valves had to be cut down, so that there was a point reached in speed where it seemed impossible to fill the cylinders with gas. Therefore, the volumetric efficiency dropped. This has been remedied to a great extent by the use of the supercharger, forcing the gas into the cylinders. The supercharger for racing car and airplane engines is having a great amount of development work done on it. Most of this is on the centrifugal type, which is generally geared to the engine with a ratio of about 4½ to 1, thus making the supercharger revolve on an engine running at 8,000 r.p.m. at a speed of 36,000 r.p.m.—some of them as a matter of fact operate faster than this.

Most of these high-speed racing engines have eight cylinders. This has helped some, for by increasing the number of cylinders it has allowed the bore to be made smaller and has assisted in cutting down the weight of the pistons and other reciprocating parts. The 1912 500-mile winner at Indianapolis had four cylinders with a bore of five inches, whereas the modern speed creations have eight cylinders with a bore of little more than two inches.

The ignition for a time was the limiting factor in engine speed, but this has been developed to such an extent that it is now keeping pace with the other developments. An eight-cylinder engine turning over 8,000 r.p.m. would have to have 32,000 sparks per minute, or if two sparkplugs per cylinder were used 64,000 sparks per minute, which is a great number for any magneto or battery system.

Before the advent of the supercharger for racing car engines, difficulty was experienced in getting proper carburetion, and some eight-cylinder engines had as many as eight carburetors on them, one for each cylinder. The use of the supercharger changed this, for not only does it compress the charge and mix it, but distributes it so well to the different cylinders that now one, or at most two, carburetors are used.

Factors Limiting Aero-Engine Speed.—The limiting factor to the reduction of displacement as well as to the speed of airplane engines is undoubtedly the loss or reduction of propeller efficiency at high speeds. This means that very-high-speed engines must drive geared down propellers, which always involves added weight and loss of power in the reduction gearing, while speeds of engines driving the propeller at crankshaft speed are, of course, limited by the air screw. Modern engines have been designed that will turn directly connected propellers at 2,200 to 2,400 r.p.m. when used in pursuit plane work. Ordinarily, speeds range from 1,400 to 1,800 r.p.m. to secure best air-screw efficiency.

QUESTIONS FOR REVIEW

1. Name some important factors influencing engine efficiency.
2. Define theoretical efficiency; actual heat efficiency.
3. What is mechanical efficiency of an engine?
4. Outline heat distribution values in high-speed engines.
5. What is the heat loss in wall cooling?
6. Name easiest method of improving engine performance.
7. What is the effect of compression increase on power developed?
8. What value has an indicator card?
9. How do cards of gasoline and Diesel engines differ?
10. What are the factors limiting compression of charge prior to ignition?
11. What is the most efficient form of combustion-chamber?
12. What effect does high speed have on engine power?
13. Why do racing automobiles use much faster engines than aircraft?
14. Does supercharging make higher speeds possible?
15. What is preignition?

CHAPTER V

TESTING AND MEASURING ENGINE POWER

Measuring Heat Engine Efficiency—Influence and Nature of Detonation—The Indicator and its Work—Manograph and its Use—High Speed Engine Indicators—Operation of Micro-Indicator—Optical Indicators—Sampling Valve Indicator—Carbon Pile Rheostat Indicator—Indicator Cards Useful—Determination of Engine Power—Indicated Horsepower—Horsepower Computations—Engine Testing Methods—How Power Curves Are Made—Simple Fan Dynamometer—Electrical Dynamometers for Motor Testing—Water Brakes and Other Tests. The Froude Dynamometer Type D.P.X.—Torque Meters—Testing Aircraft Engines—Heenan-Fell Air Brake Dynamometers—S.A.E. Engine Testing Procedure—Metric Conversion Tables.

Measuring Heat Engine Efficiency.—Mr. Harry R. Ricardo, in a series of lectures delivered at Kings College, London, and reported in the *Automobile Engineer,* states that he believes that the efficiency of internal combustion engines is best measured by air consumption rather than considering heat value of fuel supplied. The efficiency of the internal-combustion engine, unlike that of the steam-engine, does not increase with size. It may be of interest to state that though gas-engines are built in sizes ranging up to over 50 in. cylinder diameter, yet the highest efficiency so far recorded has actually been obtained with a gas-engine of only 4½ in. cylinder diameter or bore. It is well to keep in mind that in the constant volume type of internal-combustion engine, air is the true working medium. Unless the air be fully or almost fully saturated with fuel the latter will not burn with sufficient rapidity while if there be an excess of fuel present it effects the temperature but little, since it is, of course, the amount of oxygen present, rather than of fuel, which controls the temperature. In other words, neither change of mixture strength nor of throttle opening can influence the flame temperature, and, therefore, the efficiency to any appreciable extent. For a maximum flame temperature of 2,500 deg. C. the ideal limiting thermal efficiency of the cycle may be taken as approximately 80 per cent of the air cycle efficiency. On account, however, of the influence of dissociation, which is affected by pressure as well as by temperature, the true ideal efficiency approaches rather more nearly to the air cycle as the compression ratio increases.

Before proceeding further, it will be well to emphasize what is here meant by the much abused term "efficiency." It is customary to reckon the efficiency of any heat engine from the heat value of the fuel supplied, and since it is the fuel alone, and not the air, which is of commercial value, this is, of course, the practical aspect. In the cases of the constant volume internal-combustion engine, and more especially when it is using a liquid fuel, a much more accurate determination of the true efficiency can, however, be obtained from the consumption of air, since every pound of air will, by the combination of its oxygen with the fuel, liberate a definite amount of heat, whether it be saturated or super-saturated with fuel.

This is of more than academic interest for the air consumption of an engine is a true indication of its efficiency as a heat engine, while the fuel

consumption may indicate merely a waste of fuel through no fault of the engine. When the load on an engine is reduced by throttling it can be shown that so long as the mixture is constant, the indicated thermal efficiency remains the same as at full load. The popular belief that the inherent efficiency is less at reduced loads owing to reduced compression is quite incorrect for the ratio both of compression and expansion is unaltered. It should remain independent of load and in practice it does so remain except for a very slight increase in relative heat loss (provided that the mixture is the same and the ignition properly timed). The efficiency reckoned on the brake-horsepower will fall as the load is reduced owing to the larger proportion which the mechanical losses bear to the total indicated power. The efficiency of any internal-combustion engine depends upon the expansion ratio employed, the flame temperature, and the loss of heat to the cylinder walls, although this has nothing like the influence generally attributed to it.

Influence and Nature of Detonation.—When a combustible mixture of fuel and air is ignited, a nucleus of flame builds up with a rapid accéleration outwards from the point of ignition. If its rate of development exceeds a certain critical speed a detonation wave will be set up. This wave will pass through the mixture at a velocity many hundred times the normal speed of acceleration. On striking the cylinder walls the impact of this wave will give rise to a sharp ringing knock and by compressing anew the already burnt products, will still further raise their temperature until they become incandescent and actually ignite the mixture before the completion of the compression stroke.

Until recently it was always considered that detonation occurred only when the mixture was raised to a temperature in excess of its so-called ignition temperature. It is known now that this is by no means necessarily the case and that there is no such thing as a definite self-ignition temperature in the generally accepted sense. From experimental results Tizard has shown that detonation will be set up when the rate of evolution of heat exceeds the rate at which it can be disposed of to the cylinder walls. Whether the rate of evolution of heat will be sufficient to cause detonation depends upon the chemical composition and the "self-ignition temperature" of the fuel, the temperature of the flame, the temperature of the containing walls, the absolute distance the flame has to travel before it passes through the mixture, and the temperature and pressure before ignition.

The highest expansion ratio that can be employed is governed by three factors, viz., the tendency of the fuel to detonate, the surface-volume ratio (which must be kept small), and the maximum peak pressure allowable. When dealing with fuels which detonate readily, such as gasoline, the designer is restricted by the fuel to the use of a compression ratio lower than six to one, and the sale consideration is so to design the compression chamber as to permit the highest possible compression ratio to be used without detonation. To this end the maximum distance from the sparkplug to the farthest point in the combustion chamber must be kept as small as possible. At the same time the temperature of the surfaces and particularly those remote from the sparkplug must be kept as low as possible.

Detonation depends very largely upon the length of flame travel. It will be apparent that the smaller the cylinder the less the tendency to detonate and the higher the compression ratio that can be used. Mr. Ricardo claims that it is for this reason that most modern racing cars are provided with a large number of small cylinders and that power increase is best met in aviation engines by increasing the number of cylinders rather than augmenting their size, and using a smaller number. Other engineers are in full agreement with this idea and practical engines with 24 cylinders in X form are not uncommon.

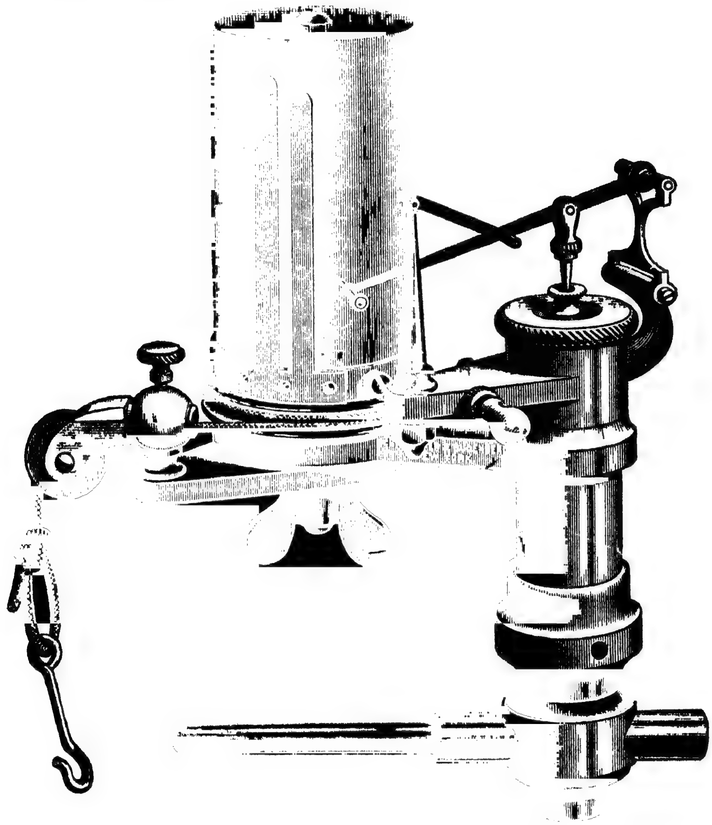


Fig. 48A.—The Thompson Indicator, an Instrument for Determining Compression and Explosion Pressure Values, and Recording them on a Chart, Suitable for Slow-Speed Internal-Combustion Engines.

The Indicator and Its Work.—The writer has selected from the many good indicators in the market one suitable for indicating the work of the explosive engine. The Thompson indicator illustrated in Figs. 48 A and 48 B, is a light and sensitive instrument with absolute rectilinear motion of the pencil, with its cylinder and piston made of a specially hard alloy which prevents the possibility of surface abrasion and insures a uniform frictionless motion of the piston. It is provided with an extra and smaller-sized cylinder and piston, suitable with a light spring for testing the suction

and the exhaust curves of explosive motors, so useful in showing the condition and proportion of valve ports. The large piston of the standard size is 0.798 inch in diameter and equal to ½ square-inch area. The small piston is 0.590 inch in diameter and equal to 0.274 square-inch area, so that a 50 or 60 spring may be used in indicating explosive engines with the small piston, which will give cards for low-explosive pressure but full enough to show the variations in all the lines. With the 100 spring and ½-inch area of piston 250 pounds pressure is about the limit of the card, but with this size piston a 120 or 160 spring is more generally used.

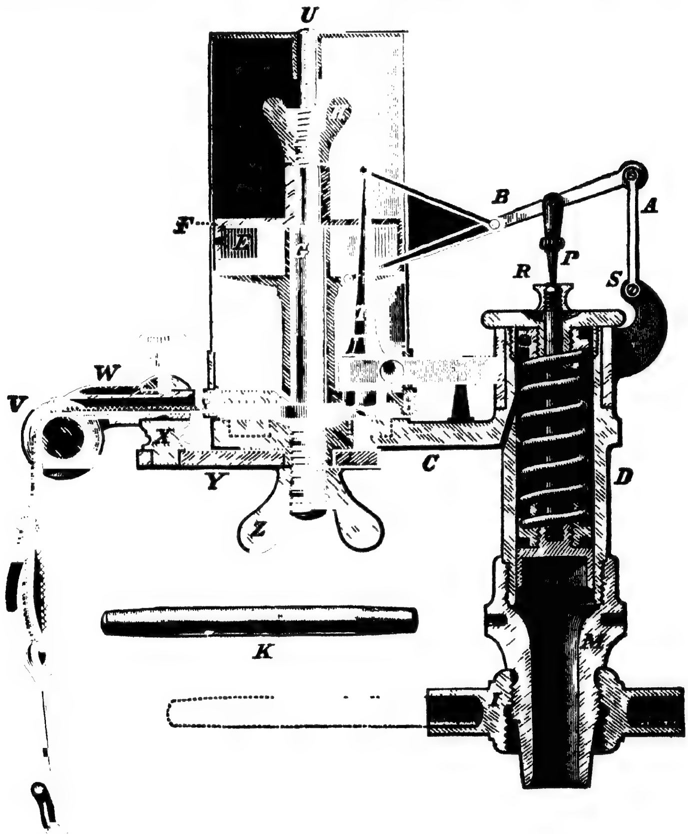


Fig. 48B.—Sectional Diagram Showing Interior Parts of Thompson Indicator.

The pulley V is carried by the swivel W, and works freely in the post X; it can be locked in any position by the small set screw. The swivel-plate Y can be swung in any direction in its plane and held firmly by the thumb-screw Z. Thus with the combination the cord can be directed in all possible directions. The link A is made as short as possible, with long double bearings at both ends to give a firm and steady support to the lever B, making it less liable to cause irregularities in the diagram when indicating high-speed motors. The paper drum is made with a closed top to preserve its accurate cylindrical form, and the top, having a journal-bearing at U in the center, compels a true concentric movement to its surface. The

spring E, and the spring-case F, are secured to the rod G by screwing the case F to a shoulder on G by means of a thumb-screw H.

To adjust the tension of the drum-spring, the drum can be easily removed, and by holding on to the spring-case E, and loosening screw H, the tension can readily be varied and adapted to any speed, to follow precisely the motion of the engine-piston. The bars of the nut I are made hollow, so as to insert a small short rod, K, which is a great convenience in unscrewing the indicator when hot. The reducing pulley is a most important adjunct of the indicator as it is necessary to make the short stroke of the indicator proportionate to the longer stroke of the engine-piston. The revolving parts should be as light as possible and are now made of aluminum for high-speed motors, with pulleys proportioned for short-stroke motors. In the use of indicators for high-compression motors it is advisable to have a stop-tube inserted in the cap-piece that holds the spring and extending down and inside the spring so as to stop the motion of the piston at the limit of the pencil motion below the top of the card.

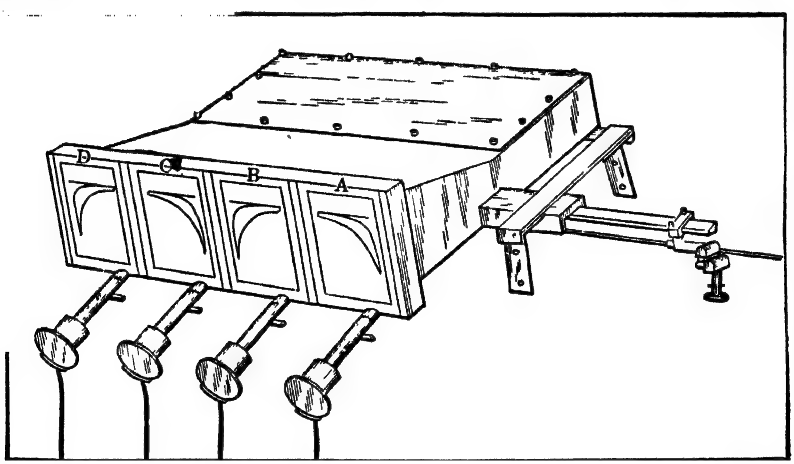


Fig. 48C.—Quadruple Manograph Apparatus for Recording the Indicator Diagrams of the Individual Cylinders of a Four-Cycle Motor, Enables à Comparison to be Made Between the Various Cylinders. The Four Glass Screens on which the Diagrams are Shown Indicated by A, B, C, D.

Manograph and Its Use.—Indicators now in use, such as the Thompson were designed originally for application to low-speed gasoline- and steam-engines but as the speed of the engine is increased so also do the imperfections of the indicator become more manifest and in the case of very high-speed engines these imperfections render the indicator quite useless. The principal defect is the number of errors due to the inertia of the relatively large mass of the moving parts of the indicator and to the friction of the joints and the indicator pencil on the cylinder. It is with a view of suppressing these defects of inertia and friction that the manograph has been designed. It is based on the principle of the deviation of a ray of light reflected by a mirror.

The indicating mechanism consists essentially of a mirror fixed at a single point and capable of oscillation in two planes. The oscillation in one

direction corresponds to the motion of the piston in the engine. The oscillations in the other plane depend upon the variations of pressure in the cylinder. A ray of light is directed onto the mirror and is then reflected by the mirror on a ground glass screen at one end of the device on which the indicator diagram is seen. On account of the lightness and efficient balancing of the mirror, it faithfully responds to every variation of pressure in the engine cylinder even at the highest engine speeds. As is true of the regular form indicator, it is necessary to transmit the motion of the engine-piston to the manograph by a system of rotary gearing and to connect it to the interior of the combustion chamber by a small water-cooled tube. The diagram appears on the glass screen in the form of an unbroken, continuous line, so the observer can follow every phase of the engine operation. Photographic film or sensitized paper may be used and permanent records obtained. With multiple manographs, one can study simultaneously the working of all cylinders. A typical multiple manograph of early design, adapted for a four-cylinder engine is shown at Fig. 48 C.

High-Speed Engine Indicators.—As the engine was developed, the rôle of the indicator became more exacting, with the result that many improvements and new instruments were devised. During early years the problem was relatively simple, because the speeds were low and the engine usually was large enough to afford a solid base upon which to mount the indicator and its parts. With the advent of the internal-combustion engine, the indicator problem became much more difficult. Compared with the steam-engine, the speeds were high and the vibration was much more intense; the small combustion-chamber and the general design made it more difficult to attach the indicator; and the very rapid pressure-changes during the cycle, as well as the high temperatures, further increased the difficulties. As the internal-combustion engine has become of such economic importance that every effort is being made to improve its performance, a satisfactory indicator should be of considerable assistance to the research engineer in his investigations of valve-timing, cam profile, sparkplug position, combustion-chamber design, fuel characteristics, and compression ratios. Used in bombs or engine combustion-chambers, it should aid the study of detonation and the action of knock inducers and suppressors.

All indicators in which the connecting linkage between the pressure element and the recording stylus or pencil is purely mechanical fall in the class of mechanical indicators. Most of them are handicapped by friction, lost motion, low natural period, and inertia. The first indicator by James Watt, in 1800, was followed by others such as the Richards, Crosby and Dobbie-McInnes, some of which have been used with considerable success on internal-combustion engines. An indicator of this type is shown at Fig. 48 A and in section at Fig. 48 B.

Operation of Micro-indicator.—The micro-indicator, designed by W. G. Collins, of the Cambridge & Paul Instrument Co., is of considerable interest. Fig. 49 A shows two views of this instrument. The engine pressure displaces a piston, having a head area of ¼ sq. in., against the elastic restoring force of a cantilever spring. The free end of the spring projects some distance beyond the point at which the piston is connected and acts as a multiplying lever, carrying a recording stylus which scratches the card on

a reciprocating sheet of celluloid. As the displacement of the stylus is several times the piston displacement, none of the usual multiplying levers are used and lost motion is thereby avoided. The natural frequency is about 1,100 cycles per second. The diagrams are less than ⅛ in. square and must be studied through a low-power microscope. Its relatively high natural frequency enables this indicator to show the disturbing effect of the air passage leading from the combustion-chamber to the pressure element. The description and illustrations are from the *S. A. E. Journal.*

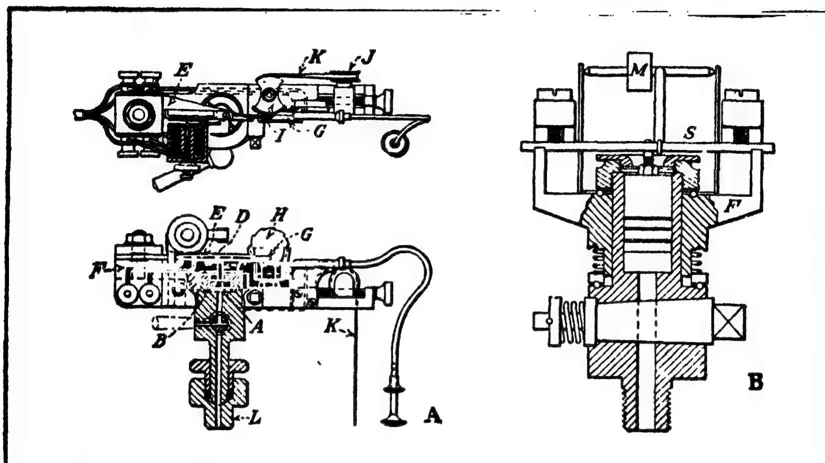


Fig. 49A.—Two Views of the Micro-Indicator. The Front End of the Cantilever Spring Carries a Stylus which can Make Minute Diagrams on the Disc of Celluloid. The Parts are: A, Cylinder Lever; B, Piston; C, Base Rod; D, Subsidiary Spring; E, Cantilever Pressure Spring; F, Frame; G, Stylus; H, Celluloid Disc; I, Cylinder for Celluloid; J, Adjustment Guide Pulley; K, Steel Disc; L, Adapter.

Fig. 49B.—Hopkinson Optical Indicator. The Frame F, is Mounted on Ball Bearings and is Given an Oscillating Motion Proportional and Corresponding to the Movement of the Engine Piston. The Movement of the Double Cantilever Spring S Causes the Mirror M to Tilt. A Beam of Light Reflected from the Mirror Traces the Pressure Diagram on a Photographic Film.

Optical Indicators.—As a class, optical indicators as shown at Figs. 48 C and 49 B are characterized by the use of a beam of light as the means of amplifying the motion of the pressure element, which may be either a small tight-fitting piston or a diaphragm. The beam of light, reflected from a small mirror, is given two motions, one from the instrument piston or diaphragm and the other a uniform rotation or reciprocation in phase with the engine piston. The beam impinges upon a stationary photographic plate or sensitized paper. By the use of small mirrors and a stationary photographic plate, the optical indicator is suitable for relatively high-speed work. The large amplification obtained with a beam of light acting as a lever reduces the necessary displacement of the piston or diaphragm, thereby reducing the wear and fatigue and at the same time making it possible to design instruments with relatively high natural frequencies. The mirrors must be very carefully mounted to avoid lost motion. All indicators employing a piston give trouble because of friction, leakage and vibration.

Sampling Valve Indicator.—The sampling valve indicator is an ordinary mechanical indicator connected to the engine by a tube containing a valve operated by the engine through gears. The valve is open during only a small part of each cycle, and the pressure in the expansion chamber of the indicator builds up until it equals the average pressure in the engine during the time the valve is open. By selecting gears having the proper ratio, the timing of the valve is changed progressively so that, in the course of a 1,000 or more cycles, the mechanical indicator used as a recorder measures all the pressures in one cycle. The drum of the mechanical indicator is driven from the same gears that operate the valve and may have either a slow uniform motion or a reciprocating motion following the valve as it progresses through one cycle of the engine.

This method of obtaining diagrams was originated by K. J. De Juhasz in 1919 and has been considerably improved since. The diagrams are often characterized by irregular "spiked" outlines. The irregularities probably are caused at low speeds by insufficient inertia in the recording mechanism, by leakage, and perhaps by engine vibration; and at high speeds by the irregularities of combustion combined with too light a recording mechanism, by the disturbing effects of the connecting tube, and by engine vibration. Increasing the weight of the recording piston, reducing leakage, and designing a very good rotating valve enabled Gale to use this method in obtaining composite cards that were relatively free from irregularities. In his latest paper M. De Juhasz also shows high-speed diagrams that are virtually free from them.

Professor Jacklin connects an indicator of this type to all the cylinders of an engine through a selector valve which enables him to obtain a complete set of cards in a short time. The composite diagram obtained with such an indicator should yield useful information if properly interpreted. When adapted to record simultaneously the composite diagrams from the several cylinders of a multi-cylinder engine, such an instrument should afford a ready and fair means of comparing pressures.

Carbon Pile Rheostat Indicator.—The new electrical indicator which will now be described was developed by the General Motors Corporation laboratory by taking advantage of work done at the Bureau of Standards by Burton McCollum and O. S. Peters, who developed an instrument for recording electrically the strains in the members of bridges, buildings, ships, and other structures. In their instrument, described in 1923, the strain to be measured causes a variation in the longitudinal pressure upon two carbon-pile rheostats, thereby changing the electrical resistance. The strain is communicated to the rheostats in such a way that the resistance of the first is increased while that of the second is decreased. The recording apparatus consists of an oscillograph and a Wheatstone bridge, the former being a sensitive recording ammeter in which a beam of light from a very small mirror records the mirror movements on a photographic film. Changes in value of the current passing through the rheostats, which is influenced by the pressure against them imposed by the diaphragm of the pressure receiving instrument indicate the value of such pressures.

Total errors resulting from the combined effects of zero shift, vibration, and temperature changes usually are less than three per cent. None of the

indicators of this type used up to the present has shown inaccuracy as great as five per cent. Therefore, it seems safe to guarantee the new indicator to be accurate within five per cent.

To meet the requirements of ordinary investigations, the necessary equipment may be enumerated as follows:

(1) An oscillograph having a number of elements equal to or greater than the number of simultaneous records desired

(2) A Wheatstone bridge for each indicator-unit

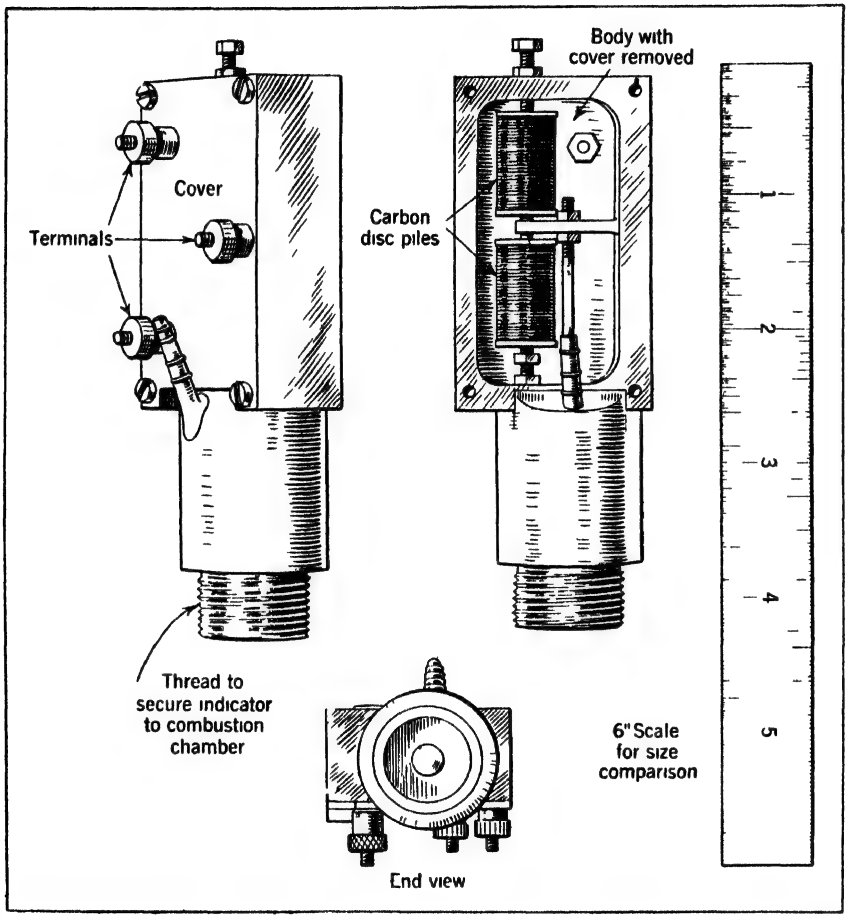


Fig. 50.—Illustrations Showing Carbon Disc Rheostat Indicator in Three Positions.

(3) If simultaneous diagrams are desired, a sufficient number of indicators having the same constants should be provided. For weak-spring diagrams, a unit should be provided having greater sensitivity than those intended for regular combustion-work

(4) Apparatus for calibration

The frame of the indicator shown at Fig. 50 is of oil-hardening tool-steel, machined so that one end is threaded to screw into the combustion-

chamber wall. In cutting out the interior of the frame, a tongue is left projecting from one side and extending nearly to the opposite side. The dimensions of the tongue are such that, when loaded with the push-rod and carbon rheostats its natural period is sufficiently high for the research for which it may be intended. The frequencies of the tongues so far constructed range between 3,500 and 500 cycles per sec. The tongue acts as a cantilever spring, the very small deflection of which is a means of measuring the pressure upon the diaphragm coupled to the pressure rod. After machining, the frame is heat-treated and then ground to final dimensions. The carbon piles are held between the tongue and the ends of the frame by pressure on plates which is controlled by the adjusting screws. The rounded points of the screws fit into recesses in the plates. At the adjusting-screw ends of the carbon piles are copper plates to which wires are soldered for electrical connections to the Wheatstone bridge. Between these copper plates and the steel pressure-plates are mica insulating-discs. The ends of the carbon piles nearest the tongue press directly against the steel plates, projections from which fit into recesses on either side of the tongue. With this arrangement, the inner ends of the piles are grounded so that the frame of the instrument can be connected to the third wire, leading to the recording instrument and the Wheatstone bridge.

The three wires are the only connections between the recording apparatus and the indicator proper, and they can be incorporated in a three-wire cable 20 to 50 ft. in length. Because of this feature, the recording apparatus need not be subjected to engine vibration. The connecting link between the diaphragm and the tongue is a hollow rod made of invar steel to avoid temperature effects. The invar rod expands so slightly that the thin diaphragm is accommodated easily to it without producing an appreciable upward force against the tongue. An invar rod three inches long expands only 0.0006 in. for a temperature change of 400 deg. Fahr. To produce the same displacement of the diaphrams generally used in indicators of this type, a force of 1.5 lb. would be required. In practice, the rod becomes heated at one end only, being cooled by the stream of air directed against the diaphragm.

The diaphragm that receives the pressure in the interior of the cylinder is a thin steel disc brazed to the threaded end of the frame and connected at its center to the push-rod. The elastic constants of the diaphragm have no influence upon the records obtained. So great is the sensitivity of the carbon piles, combined with the recording apparatus and the design of the tongue, that so far no work has been done in which the deflection of the diaphragm has equalled 0.0005 in. Operation of this instrument does not depend upon an elastic diaphragm, because the displacements are so small that the restoring forces set up in the diaphragm are negligible compared with those set up in the tongue, which is located where it remains cool. The diaphragm is made of a tough spring-steel sufficiently thick to withstand engine pressures. In designing the instrument to meet special requirements, the dimensions of the tongue and diaphragm are so proportioned that, for a given displacement, the restoring force created by the diaphragm displacement is negligible compared with that of the tongue. The life of the diaphragm is prolonged by directing a stream of air against

it, through the side tube shown in Fig. 50 which shows the indicator from different angles. A very complete description of this indicator, from which the preceding extracts have been taken will be found in the *S. A. E. Journal* for July, 1928. The most obvious shortcoming of the indicator in its present stage of development is that no means are provided for taking pressure-volume cards. For fundamental research, the pressure-time cards are probably much to be preferred; but for routine work, pressure-volume cards also are desirable without the trouble of re-plotting.

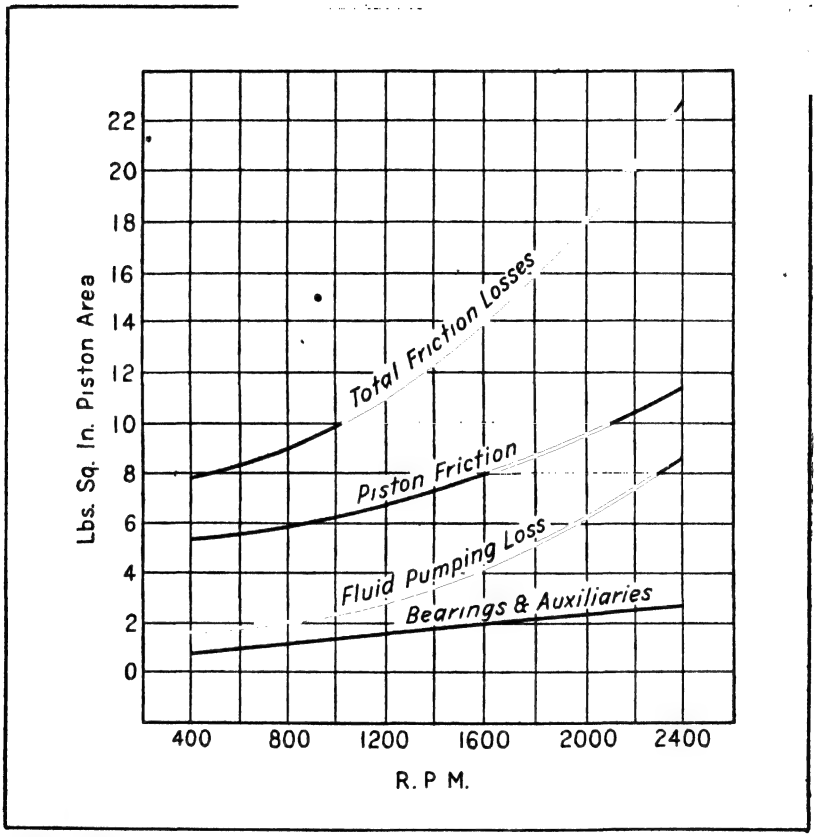


Fig. 50A.—Diagram Showing Power Losses Due to Friction in Internal-Combustion Engines at Various Speeds and Pressures.

Indicator Cards Useful.—If indicator cards are available, many things can be seen directly, whereas without the indicator it is necessary to use indirect methods to ascertain the effect of various factors. The change of an operating condition such as speed, air density or mixture ratio, affects the indicated power liberated by combustion inside the cylinder in a direct manner; but it may or may not change the power required to overcome friction losses. For example, a change of mixture ratio sufficient to decrease the brake power has probably not materially changed the friction, if the speed and air density remain constant, but a change of air density

at constant speed and with a suitable mixture ratio will change not only the indicated power but the friction power as well, and there will be at least two causes for change of brake power. In other words, when studying a change of operating conditions, the effect upon indicated power is the fundamental relation and the effect upon brake power is only a most important by-product.

Friction Loss Diagram.—The friction losses are always a promising field for investigation, not only because they must be understood in order to be reduced but because, at present, we must know their total amount to obtain a measure of the energy liberated inside of the cylinder. The term friction as here used includes not only the piston and bearing friction but also the power required to operate the valves, pumps, ignition devices, etc., and the pumping losses to charge and scavenge the cylinder. It would appear that the friction as measured when the engine is being driven by an electric motor might not be the same as the friction when the engine is operating, because of the effect of the explosion pressures and temperatures, the effect of the fuel upon the cylinder-wall lubrication, and other features. The values of indicated power obtained by adding the brake and friction power appear to be a good measure of what would be obtained with an indicator. The diagram at Fig. 50 A shows friction losses in a test at various speeds and pressures.

Determination of Engine Power.—Many readers of inquiring mind may have tried to understand the methods of rating internal-combustion powerplants the vogue and in making inquiries to satisfy a natural curiosity, or looking up authorities on mechanical subjects, have been confronted with such a mass of technical data that their efforts to enlighten themselves have been in vain. Yet the question of power determination is a vital one to every engine user, as it is the capacity of the engine that determines the speed and climbing ability when used in an airplane, or its capability for operating automobiles or other machinery when used in automotive or stationary applications.

Power is not hard to define in simple language, and the various dynamometers and other testing machines used to determine its value are not hard to understand as they are based on principles that are easily explained. Power is always defined as the rate of doing work, which in turn is the product of a force acting through a certain distance in a certain time. If a man raises a weight of 50 pounds one foot, he has done work equivalent to 50 foot pounds, but unless some unit of time is stated, it is very difficult to make any comparison with any standard unit. It may have taken the man one minute to lift the load that distance; obviously he did not exert himself or work as hard as though the weight had been raised one foot in one second. The standard unit of measurement is one horsepower, which is the ability of lifting 33,000 pounds one foot in one minute or 550 pounds one foot in one second, or any combination of weight, distance and time that will produce the same resulting product. Power is not work but a measure of work.

In any gasoline or crude oil motor, which is the popular automotive powerplant of the present century, power is obtained by the rapid combustion of an inflammable gas in the cylinders. The pressure resulting from expansion tends to force a movable member, called the piston (which

works inside the cylinder) down a certain distance, called the stroke. The piston is coupled to a crank by a connecting-rod as its reciprocating movement must be converted to a rotary motion in order to utilize it more effectively in driving the traction wheels of the automobile or the rotating propeller of an airplane. Thus the amount of the expansion gives a force, the length of the stroke gives the distance through which the force is applied and the number of strokes per minute adds the time factor. If one knows the amount of the explosive force or pressure acting on the piston top in pounds, the piston travel in inches or feet and the number of strokes during which power is applied to the piston per minute the horsepower that should be theoretically obtained from a gasoline-engine cylinder can be readily approximated.

Indicated Horsepower.—Thus in determining indicated horsepower, one must consider the following cardinal points. First, the mean effective pressure against the piston top, which is an average of the number of pounds per square inch acting during the entire stroke. Second, the area of the piston top in square inches. Third, the length of the stroke in feet. Fourth, the number of explosions per minute. The product of these factors divided by 33,000 gives the approximate horsepower, and may be expressed by the following simple formula:

$$\text{I. HP.} = \frac{P \times A \times S \times \text{E. P. M.}}{33{,}000}$$

In which P is mean effective pressure in pounds per square inch.
A is area of piston top in square inches.
S is length of stroke in feet.
E. P. M. is number of explosions per minute.

Horsepower Computation.—A number of other formulæ have been proposed in which various factors are considered. Some of these, which are of English derivation follow:

Denby Marshall Formulæ:

Measurement in inches.	Measurement in mm.
$\dfrac{D^2 S N}{12}$	$\dfrac{D^2 S N}{200{,}000}$

This formula gives a close approximation of power, and assumes a revolution speed of 1,000 per minute, or with modification for effect of stroke-bore ratio on revolution speed.

$\dfrac{D^2 S N \text{ Revs.}}{12{,}000}$	$\dfrac{D^2 S N \text{ Revs.}}{200{,}000{,}000}$

Institution of Automobile Engineers' Formulæ:

Measurement in inches.	Measurement in mm.
$.45\,(D + S)\;(D - 1.18)\,N.$	$\dfrac{(D + S)\;(D - 29.97)\,N}{1{,}433}$

This formula embodies a correction for mean effective pressure rising with bore, and one for effect of stroke-bore ratio on speed.

Lanchester's Formulæ:

Measurement in inches.	Measurement in mm.
$.5\,D^2 N\sqrt{R}$	$\frac{D^2 N}{1.290}\sqrt{R}$

This formula provides a correction for piston speed limited by the stroke.

Burl's Maximum Rating Formulæ:

Measurement in inches.	Measurement in mm.
$\frac{1}{2} D\,(D - 1.18)\sqrt{\frac{D^2 S}{M}}$	$\frac{D}{1{,}290}\left(D - 29.97\right)\sqrt{\frac{D^2 S}{16{,}390\,M}}$

This formula is similar to the I.A.E. formula, but embodies a speed limiting factor based on weight of reciprocating parts, and M = mass of weight of reciprocating parts in one cylinder. For cast-iron pistons

$$M = .08\,D^3\,(1 + .15\,R) + 1.5 \text{ lb.} \qquad M = \frac{D^3}{204{,}700}\left(1 + 15\,R\right) + 1.5 \text{ lb.}$$

Poppe's Formulæ:

Measurement in inches.	Measurement in mm.
$\frac{D\,S\,N}{2.5}$	$\frac{D\,S\,N}{1{,}612}$

This formula is based upon cubic capacity of cylinders simply.

Engine Testing Methods.—This discussion of engine efficiency would not be complete without reference to engine-testing equipment which has been an essential factor in the development of the internal-combustion engine used in automobiles and airplanes. There are two distinct methods of testing an engine; one, a dynamometer test and the other, a test after installation in the car, truck, tractor, plane or boat, as the case may be. Both kinds of test are necessary, each method permitting of acquiring certain data indispensable to a complete understanding of the subject. The dynamometer represents a means of absorbing power and may consist of a prony friction brake, a water brake, a fan brake or an electric-cradle dynamometer. A typical cradle dynamometer is shown at Fig. 51 B.

A practical electric dynamometer installation consists of a generator, adapted also to run as a motor. The field frame of this generator is mounted on ball bearings and is carefully balanced. The engine to be tested is connected by a suitable coupling to the armature shaft and the field is separately excited. The current generated is dissipated in suitable resistance grids and no electrical readings are taken. The load on the engine is varied by a suitable field rheostat and, in addition, the resistance grids can be connected in various groups to obtain the desired load. The actual load on the engine is represented by the torque on the field frame of the generator, and this is measured by suitable scales such as are shown in Fig. 51 C on the near side of the dynamometer. When making a power test, therefore, the torque readings are given directly by the scales and the speed readings are obtained by suitable tachometers, preferably by a positively-driven revolution counter.

How Power Curves Are Made.—A typical power curve of a gasoline-engine is shown in Fig. 51 D. It will be noted that the torque is plotted against the ordinates shown on the right-hand side, which give the pounds pull at 21-in. radius, this being the distance from the center of the field frame to the point where the weighing scales are attached. The revolutions per minute are plotted against the abscissas and the brake-horsepower is obtained by multiplying the number of revolutions per minute by the pounds torque and dividing by 3,000. The brake mean effective pressure is given in pounds per square inch and is obtained by multiplying the number of

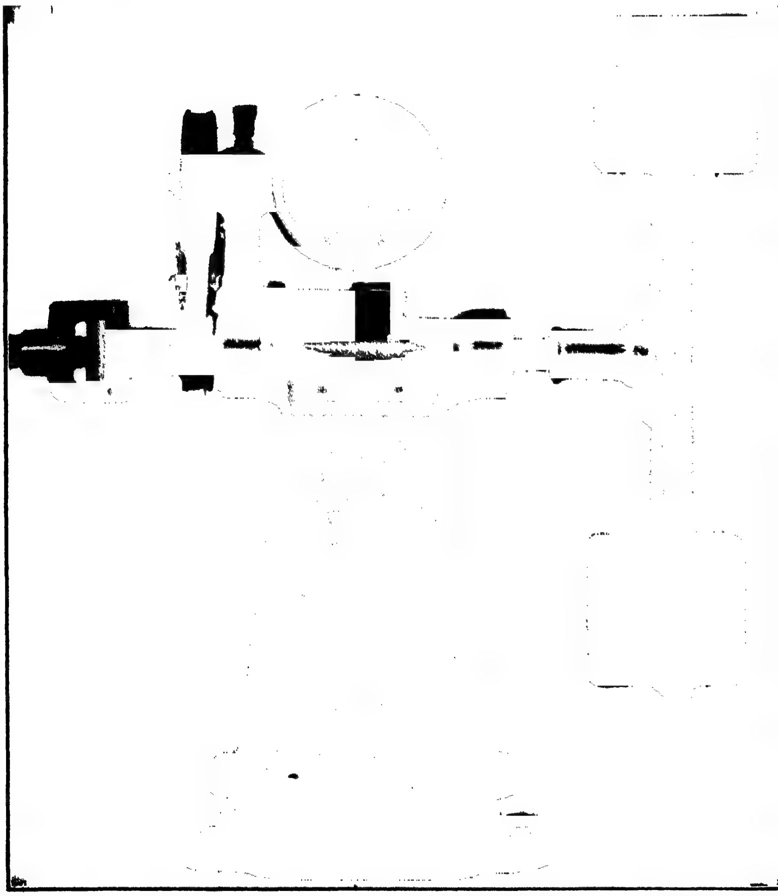


Fig. 51.—Typical Fan Dynamometer for Making Horsepower Determinations.

pounds torque by a factor varying with the displacement of the engine. Gasoline consumption is plotted on a basis of pounds of gasoline per horsepower-hour. In actual testing the gasoline consumption is obtained by the difference in weight of the gasoline supply tank before and after a run, which may vary from one to fifteen minutes, or longer, depending upon the degree of accuracy desired. The usual procedure is to obtain the required data over the entire speed range of the engine in a series of steps, the speed increase in successive runs being about 200 r.p.m. The radius of the

arm attached to the weighing scales will vary in different testing machines, the longer the arm is or the greater the distance from the center of the field frame to the scale, the less the amount of pull indicated for the same power.

Simple Fan Dynamometer.—Many manufacturers who are producing engines in large quantities regard a running-in test as one that has sufficient value for all practical purposes and as a simple fan dynamometer may be easily attached to the powerplant, it is not difficult to form an opinion of the capacity of the motor by comparing its action with that of others known to be efficient. The fan dynamometer is very simple and economical. A typical appliance of this kind is shown at Fig. 51, it consisting essentially of a through shaft supported on ball bearings mounted in a heavy cast iron standard or base, having a wooden arm carrying two aluminum plates at one end and a universal joint or other coupling for attaching to the engine to be tested on the other. In this instance, a tachometer or revolution indicator is fitted, this being driven by the shaft carrying the resistance arm and plates so the speed of the fan may be easily determined. Horsepower determinations are made by calibrating the fan from some standard motor and checking the results by some other form of dynamometer to determine if the fan is correct at the start. When once set, however, the power delivered by any motor may be noted by ascertaining the number of revolutions per unit time and comparing the results with a plotted curve or chart determined by a former calibration.

The fan dynamometer is usually employed to note the effect of a maximum load as it must be run sufficiently fast so the resistance will impose a steady load upon the motor to be tested. It is not practical to run such a device at low speeds because the load cannot be varied as easily as in a Prony brake or electric dynamometer. To vary the resistance offered by such an appliance, the vanes must be shifted on the arm to which they are attached. The nearer the center they are placed, the less the resistance offered and the higher the speed possible with the same motive force. When one desires to test the output of a motor under varying speeds, such as might be caused by varying the mixture quality or quantity or spark time, and also to determine the capacity of the engine at low speed, the fan dynamometer has disadvantages which militate against its use. The principle of air resistance is made use of in apparatus of this type. It has been determined by Kempe that the power required to move a plane surface through the air varies with the velocity, it increasing as the cube of the speed.

In the case of aviation engines, the propeller the engine should drive is a good index of its power output and running-in and test stands similar to that shown at Fig. 51 A, are practical and give good results in testing engines after overhauling. If the engine attains or exceeds its rated r.p.m. pulling a standard screw it is evidence that the engine is suitable for flight work and that the repairs have been properly executed.

Electrical Dynamometers for Motor Testing.—When one demands a flexible dynamometer, it is difficult to conceive any apparatus that is superior to those in which electricity is utilized. Electric testing apparatus may be either of two classes: that in which a standard dynamo electric

machine is driven by the motor to be tested and its output determined by suitable resistance and measuring meters, and the other form having an oscillating field member restrained from undue movement by a lever arm and weights. The dynamo is attached to the engine shaft by a flexible coupling and its output measured by suitable instruments on the switchboard. The current developed is absorbed by resistance coils and in some plants where engines are continually on test, the current may be utilized

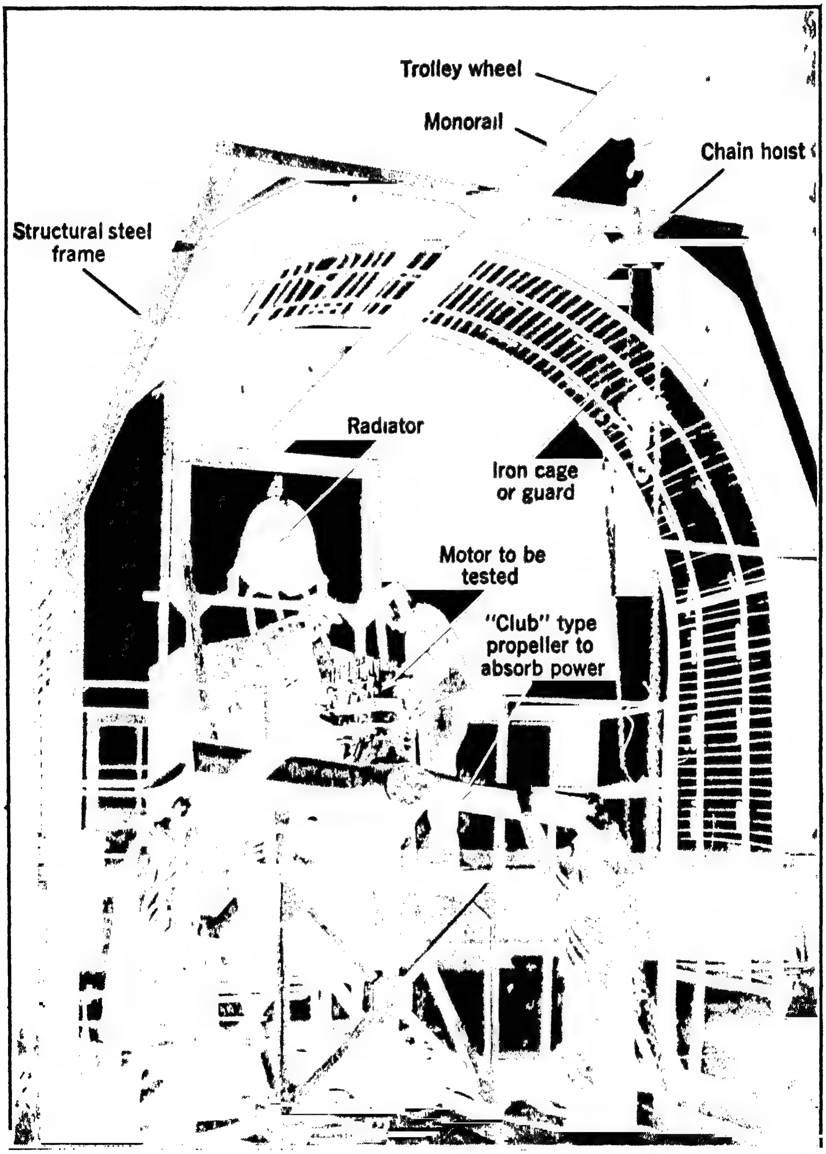


Fig. 51A.—Test Stand for Determining Horsepower of Aviation Engines by Using Propellers of "Club" Type to Absorb Power Generated.

by the electric motors employed in operating the shop machinery. A tachometer is attached to the wall just above the dynamo and is driven by a pulley on the end of the dynamo shaft. As the amount of current delivered will vary with the speed at which the dynamo armature revolves, and this in turn is dependent upon the power of the engine, it will be evident that the power can be read directly from the current recording instruments. Electrical horsepower is measured in watts, it taking a little over one mechanical horsepower to furnish 746 watts of current. This unit is obtained by multiplying the amount of current as expressed in amperes by its pressure in volts; thus, if the voltage was 100, it would take 74.6 amperes of current to do work equivalent to an electrical horsepower. In other words one may say that 746 watts is equal to 33,000 foot pounds per minute.

Suppose the dynamo coupled to the engine tested was delivering a current of 7,460 watts or nearly 7.5 kilowatts, if there were no losses in transforming mechanical into electrical energy, the engine would be developing ten horsepower. As it is, the power exerted by the powerplant would be about ten per cent greater than this, or eleven horsepower. If the indicating instruments showed that a greater current of 37.5 kilowatts was being absorbed by the resistance grids or power lines, the gasoline engine would be delivering in excess of 55 horsepower.

Sometimes a bank of incandescent lamps is utilized instead of the resistance coils for absorbing the current produced, in other installations the electrical energy is made to do useful work by operating machinery, or it may be fed direct to lamps and other apparatus. The current is sometimes conserved by means of a storage battery installation, and in this way the fuel used in running the engine is not wasted. In addition to routine tests such as outlined, the electric dynamometer permits much other information to be obtained. For instance, the dynamometer can be used as an electric motor to turn over the engine and, by taking the readings of the torque required and the revolutions per minute, a friction-horsepower curve can be arrived at, giving the mechanical efficiency of the engine under these conditions. This information, however, is of comparative value only, since the actual mechanical losses are very much greater when the engine is running under its own power due to the gas pressures. The effect of carburetor settings, ignition and valve-timing, and many other variables can be investigated to the best advantage on the dynamometer.

The class of electro-dynamometer shown at Fig. 51 B differs from the dynamo test previously described in that a lever arm is attached to the field piece of the machine, which is journaled in a frame in such a manner that the tendency of the field is to follow the armature, this pull increasing as the horsepower is augmented just as in Prony brake. The weights at the end of the long lever resist the tendency of the field to follow the armature. Suitable electrical instruments are mounted on the board to control the current, and the horsepower delivered by the motor may be easily computed by a simple formula very similar to that previously mentioned for measurement of torque absorbed by friction brakes. The cradle dynamometer, as the form shown at Fig. 51 C is called, is often used in making engine tests with various auxiliary systems to determine relative efficiency of carburetors, ignition systems, etc. This is because of the close gradua-

tion possible of the load and accurate determination of power delivered. The apparatus used in determining fuel consumption in connection with electrical or other dynamometers is simple. The fuel container is placed on a platform scale and the amount consumed for any given period may be easily determined by the diminution in the aggregate weight. Air consumption can also be measured as well as lubricating oil used.

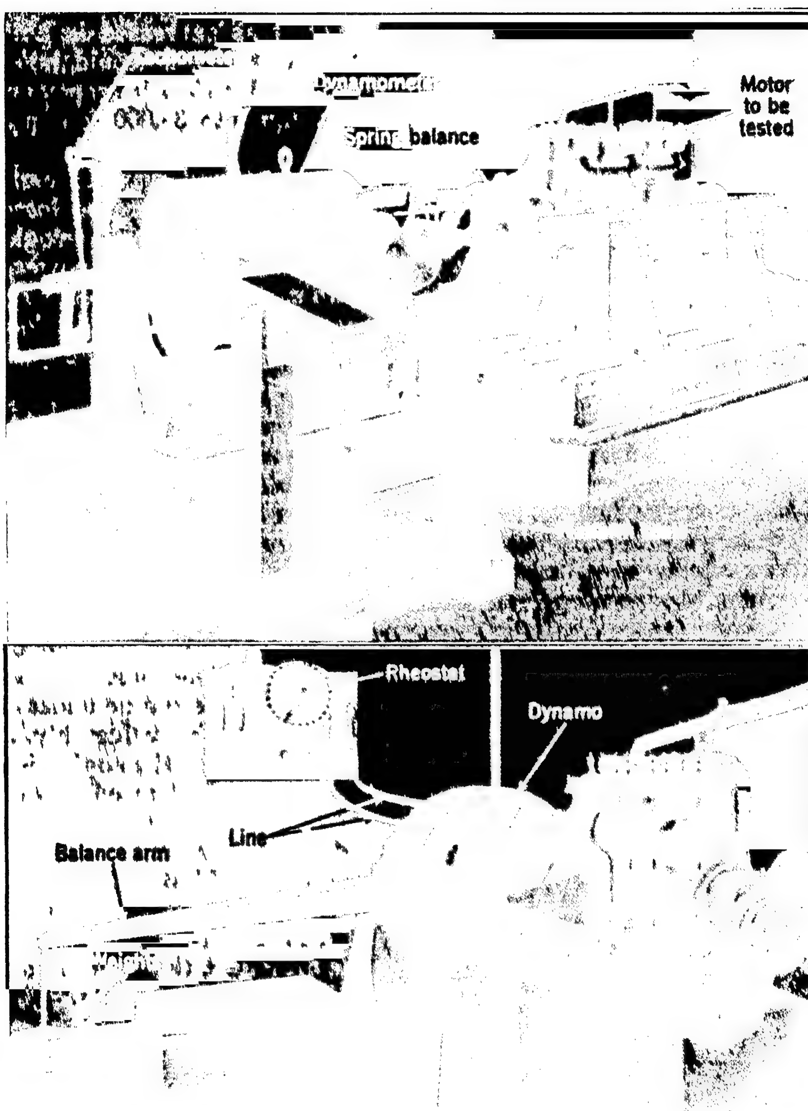


Fig. 51 B and C.—Lower View B Shows Simple Form of Electric Cradle Dynamometer. Another Type of Electrical Cradle Dynamometer is Shown at C.

Water Brakes and Other Tests.—In addition to the common methods outlined, some engineers have devised testing apparatus in which water or other liquids play a part, the power being absorbed by some form of circulating pump or paddles in liquid, and its quantity measured by determining rate of flow or revolutions per minutes of the paddles in the resisting medium. One form, known as the hydro-dynamometer, consists of a large disc or a plurality of such members revolved at high speeds in a water bath contained in a suitable housing. Others are merely large centrifugal pumps, and still others consist of an ordinary large boat propeller revolved in a tank of brine or oil.

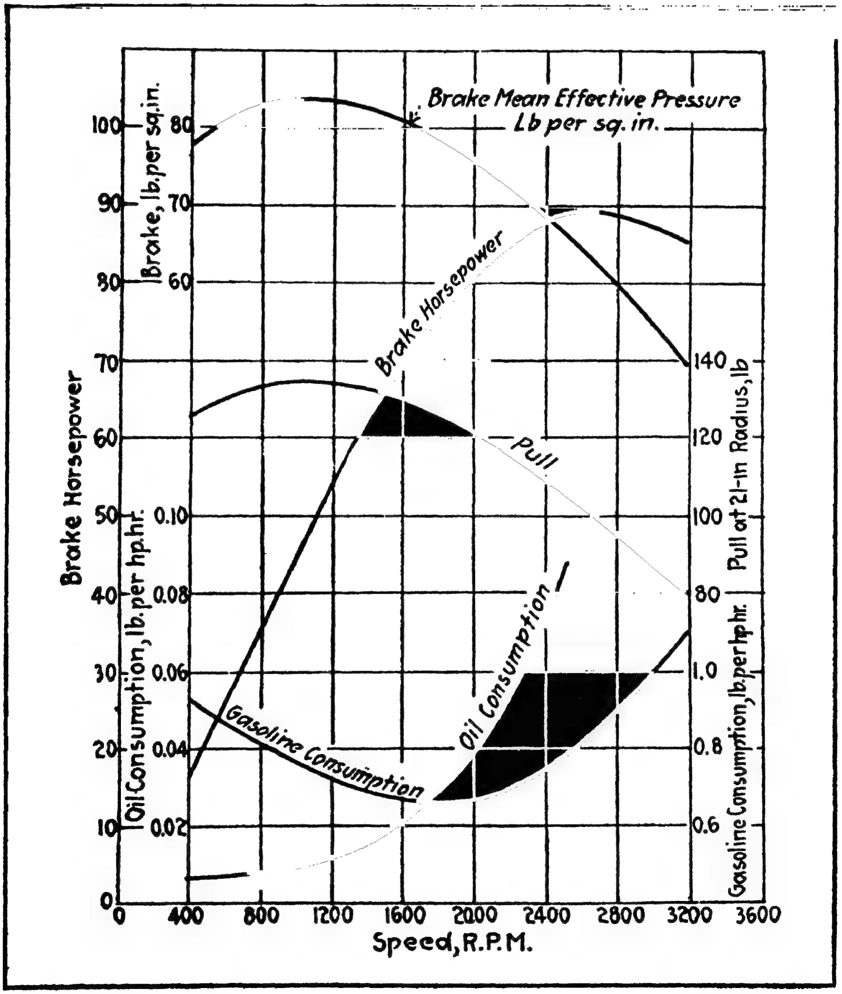
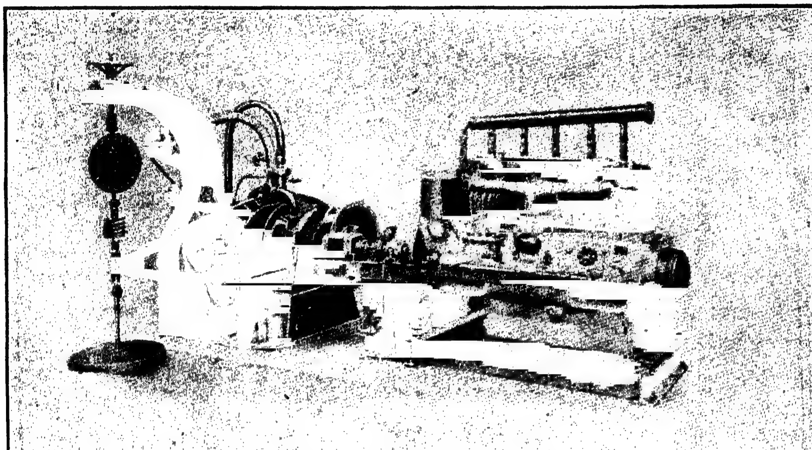


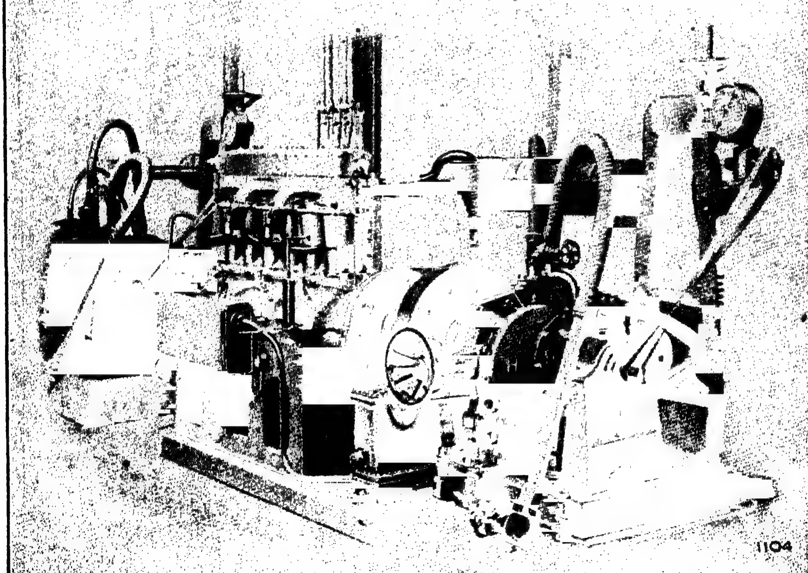
Fig. 51D.—Typical Diagram Showing Engine Performance as Determined by Dynamometer. Note Completeness of the Information Given by Graphic Charts of this Character, which are Invaluable to Engine Designers.

All aviation engine factories have some form of testing apparatus to make sure that the engines produced have the proper power before they

are shipped, and after receiving a thorough test, they are sent to the purchaser with every assurance that they will function properly and deliver full power as long as operating conditions are normal. The length of time that a motor will retain its efficiency after leaving the factory depends upon many factors, one of the most important being the amount of care it receives at the hands of the owner, or those entrusted with its main-



Froude Dynamometer, Type D.P.X. in test shop of Messrs. Daimler Motor Car Co. Ltd.,



240 b.h.p. 6-Cylinder Aeroplane Engine coupled up to Froude Dynamometer. (One of some 40 such test-benches at works of Messrs. Armstrong-Siddeley Motors Ltd., Coventry).

Fig. 51E.—Types of "Froude" Water Brake Dynamometers that May be Used for Testing Horsepower of Automotive Engines and that are Capable of Continuous Operation.

tenance. The best of motors will fail in service if neglected or abused, while an indifferent design may give very satisfactory service if properly cared for.

Another class of engine testing, such as the road testing of an automobile engine or a flight test of an airplane engine, for example, requires more than ordinary intelligence on the part of the testing engineer. The personal element entering into such testing is very large. We must frequently rely on the senses of the tester as much as on the instruments to judge whether a certain change represents an improvement or not. It is of course possible to reduce this kind of testing to a science by the use of instruments such as an accelerometer, recording tachometer, thrustmeter, etc., but in field practice it is not always possible to go to the trouble of installing such instruments.

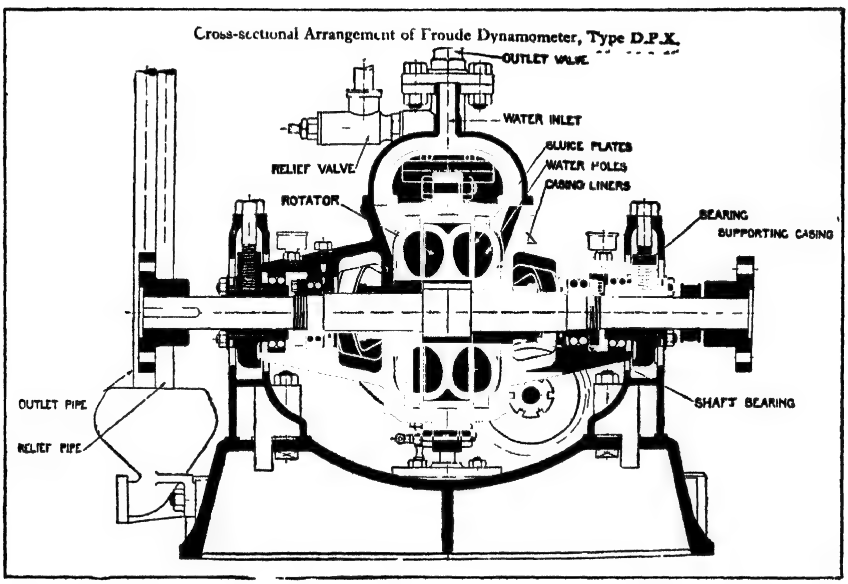


Fig. 51F.—Sectional View Showing Internal Construction of "Froude" Type D.P.X. Dynamometer. Note Simplicity of Rotating Element and Shape of Water Pockets.

The "Froude" Dynamometer Type D.P.X.—A cross-sectional drawing through the D.P.X. type "Froude" Dynamometer is given at Fig. 51 F.

The main shaft is carried by bearings fixed in the casing (not in external supports). The casing in turn is carried by antifriction trunnions, so that it is free to swivel about the same axis as the main shaft. When on test the engine is directly coupled to the main shaft, transmitting the power to a rotor revolving inside a casing, through which water is circulated to provide the hydraulic resistance and simultaneously to carry away the heat developed by destruction of power. In each face of the rotor is formed pockets of semi-elliptical cross-section divided one from another by means of oblique vanes. The internal faces of the casing are also pocketed in the same way. When in action the rotor discharges water at high speed from

its periphery into the casing pockets, by which it is then returned at diminished speed into the rotor pockets at a point near the shaft. Thus, the pockets in rotor and casing together form elliptical receptacles round which the water courses at high speed, creating vortices which destroy the power of the engine as quickly as it is developed.

Between the faces of the rotor and the casing thin metal plates or sluices are interposed, capable of sliding radially towards or away from the shaft. By rotating an external handwheel these sluices can be caused to mask or uncover more or less of the power-absorbing cups, and in this way to regulate the load exerted by the dynamometer. In this type of dynamometer under running conditions the pockets should remain always full of water which enters through holes in the casing to a point near the center of each vortex where the pressure is low. After being heated by the destruction of power, the water passes to waste through an outlet valve, which in practice is regulated to give a temperature of about 140 degrees Fahrenheit to the water.

All "Froude" dynamometers have casings supported upon antifriction trunnions and free to swivel about the axis of the shaft. The shaft is mounted upon bearings fitted into the casing and not in independent supports. The piping conveying water to and away from the casing is flexible, and precautions are taken to prevent external forces from resisting the free swivelling motion of the casing. When a turning moment is applied to the shaft, any resistance created by the dynamometer, whether by hydraulic or solid friction, must of necessity react upon the casing, which tends to swivel upon its supports. In the "Froude" dynamometer this tendency is resisted solely by a weighing machine, which correctly indicates the magnitude of the torque. This arrangement ensures scientific accuracy.

The form of weighing apparatus varies somewhat. In the D.P.X. type dynamometer it consists of an arm directly attached to the casing, and connected through pin joints to a spring balance upon which dead weights are suspended so as to put the springs of the balance into tension. When the dynamometer is at rest, uncoupled from the engine, and the arm is in a horizontal position, the pointer indicates zero. The dial is graduated so as to give a direct reading of the lifting force exerted by the end of the lever arm when at work. See Fig. 51 E.

Calculation of Power.—In any dynamometer which measures the torque developed by a prime mover by means of the force exerted at the end of a lever arm which is actually or virtually attached to the nonrotating carcase of the dynamometer, the basic formula for the calculation of brake horsepower is:

$$\text{B.Hp.} = \frac{W \times 2 W R N}{33{,}000}$$

Where

W = Net weight lifted or force exerted at end of dynamometer arm in pounds.
R = Effective radius of arm in feet.
N = R.p.m.

This formula is simplified in the "Froude" Dynamometer by making R of suitable magnitude, and becomes

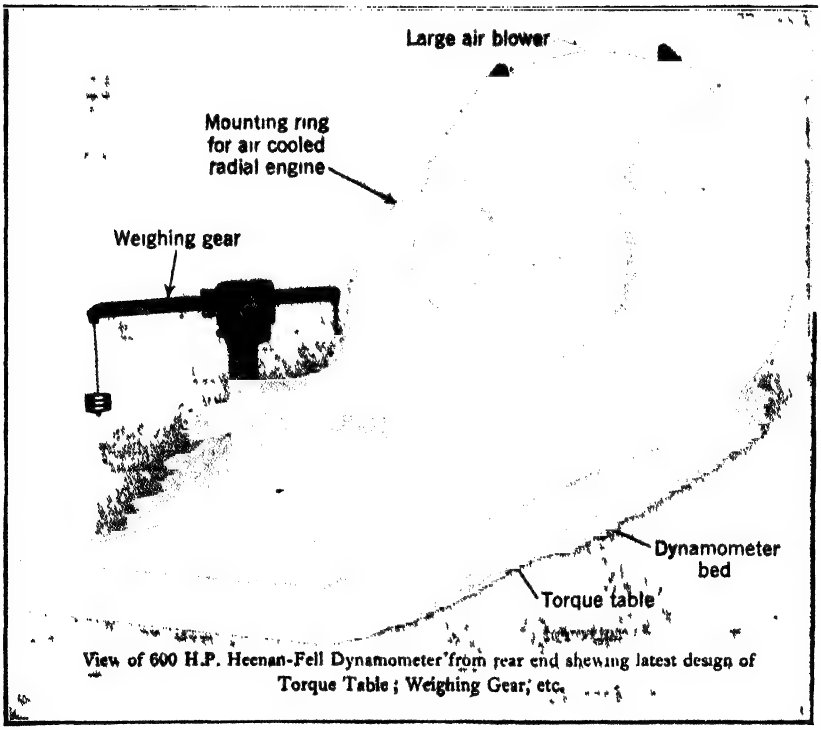
Thus, if

$$B.Hp. = \frac{W\,N}{K}$$

$$R = 5.2521 \text{ feet.}$$

$$B.Hp. = \frac{W\,N}{1{,}000}$$

In a dynamometer built to give Metric units of measurements, a similar plan is followed. Thus, in a Metric machine having an arm 1,432.4 millimeters long, $K = 500$.



View of 600 H.P. Heenan-Fell Dynamometer from rear end shewing latest design of Torque Table; Weighing Gear; etc.

Fig. 51G.—View of Heenan-Fell Dynamometer Which Uses Air Fan to Absorb Power and to Cool Engine. Note Engine Mount for Radial Air-Cooled Engine.

Torque Meters.—Torque reaction has hitherto been measured generally by suspending dead weights upon, or attaching a weighing machine to, a lever arm extending from the swivelling carcase of the dynamometer; an illustration of this system is shown at Fig. 51 E. The space occupied by such an arrangement is necessarily considerable, and the mass of the dead weights necessary to resist the very heavy torque developed by modern Prime movers renders adjustment somewhat cumbersome. The new S &

F type "Froude" dynamometers incorporate a Torque Meter in which these defects are absent, and of which an arrangement is shown at Fig. 51 H. The casing, supported on antifriction trunnions not shown in the diagram, is provided with two lugs at diametrically opposite points on the periphery; each lug is coupled up by a connecting rod having articulated joints to a system of levers fixed in the base-plate.

When the direction of rotation is counter-clockwise as shown, the left hand connecting rod presses downwardly upon the short arm of a main lever, pivoted upon a fulcrum which is connected by a rocking pin joint to the bedplate. The right hand connecting rod pulls vertically upwards

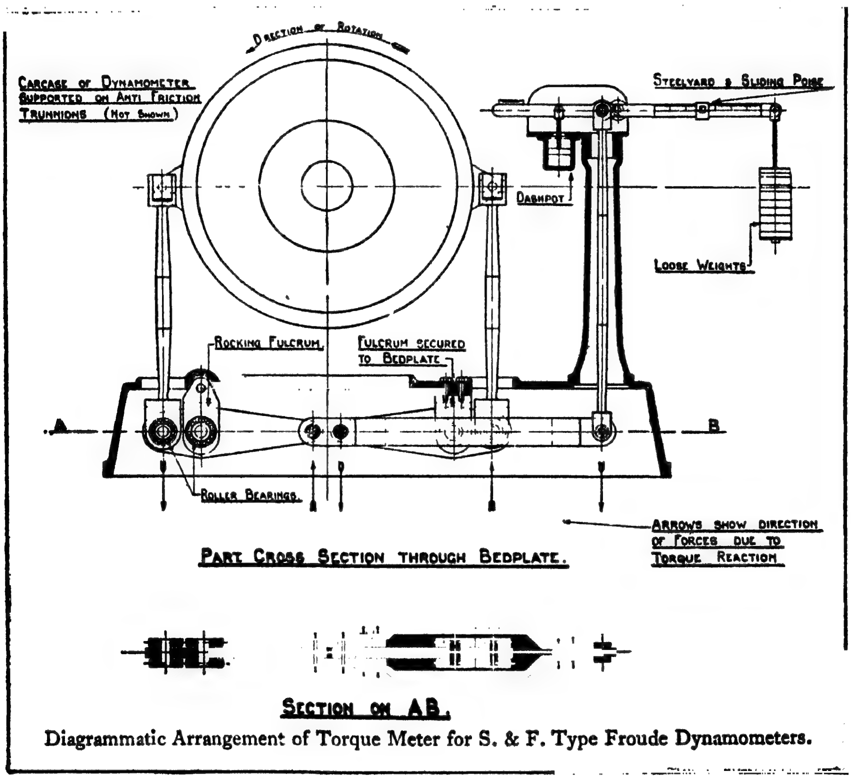
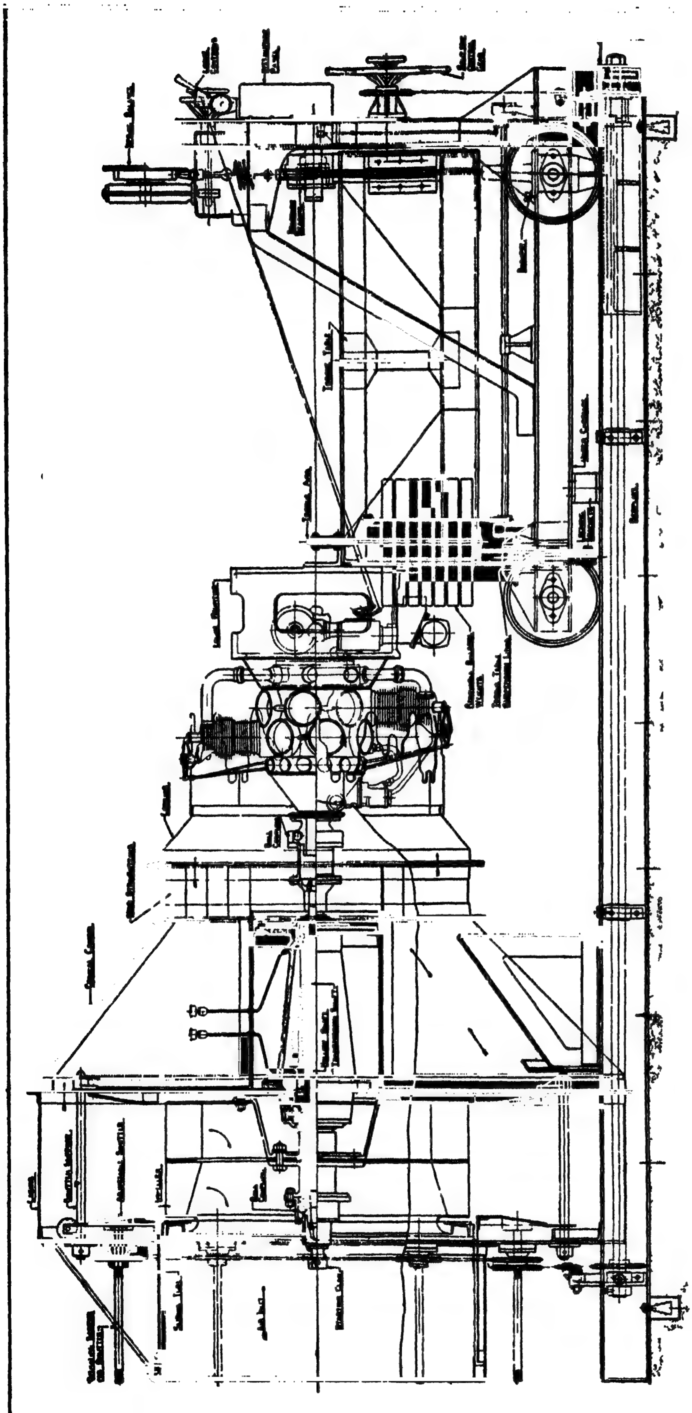


Fig. 51H.—Sectional View Showing Arrangement of Torque Meter for S and F Type "Froude" Dynamometers.

upon the short arm of a similar main lever, which is pivoted upon a fulcrum permanently fixed to the bedplate. The long arms of the main levers, therefore, exert respectively upward and downward forces, and their ends are connected together by a third lever transmitting the resultant of these two forces to the short arm of a steelyard type of weighing machine. This system of levers is equivalent in effect to a lever arm directly attached to the dynamometer casing, and having an effective length several hundred times as great as the radius from center of shaft to center of one of the casing lugs. The torque can therefore be measured by a number of very small weights and a sliding poise instead of enormously heavier weights.



Cross-Sectional Arrangement of Heenan-Fell Air Brake Dynamometer.

Fig. 51-I.—Diagram Showing Complete Installation of Heenan-Fell Air Brake Dynamometer and Radial Air-Cooled Engine Mounted Thereon for Testing or Running In.

Testing Aircraft Engines.—Tests upon aircraft engines can only be conducted with the aid of special equipment of a somewhat more elaborate nature than that which suffices for testing the majority of motor-car engines. The ventilation of such engines, for example, especially those which are air cooled, or being water cooled are fitted with turbine driven superchargers, must be carried out under conditions analogous to those experienced in flight; and this requirement involves the projection of a powerful blast of air against the heat-radiating parts of the engine. Until recently the air blast has usually been supplied by powerful centrifugal fans driven by electric motors of size up to several hundred horsepower, of which both the first cost and running cost are extremely heavy. Moreover, in many cases such as in mobile military establishments, neither the necessary electric supply for driving the motors nor the water supply for the "Froude" dynamometer is available.

To remedy these deficiencies the Heenan-Fell Air Brake dynamometer has been introduced, of which a reproduction is shown at Fig. 51 G. In principle this dynamometer converts the power of the engine into a powerful air-blast which is blown over the heat-radiating parts of the engine; while at the same time it measures the reactionary torque upon the crankcase.

The Heenan-Fell Air Brake Dynamometer.—A cross-sectional elevation of the above dynamometer, with an air-cooled radial engine fitted in readiness for testing is shown at Fig. 51 I. The engine is carried upon a Torque Table which is pivoted upon link gear so that it is free to oscillate about the axis of the propeller shaft. This table is connected through an extended lever arm to weighing apparatus by which the reactionary torque and hence the horsepower can be accurately measured. A cardan shaft transmits the power to a single inlet centrifugal fan impeller of special design, capable of exhausting air from the surrounding atmosphere and blowing it under pressure into a heavily built casing having an axial discharge opening. The impeller is surrounded on the periphery by a large sliding tube the inner end of which carries a dished flange, of which a duplicate of opposite handing is permanently fixed to the interior of the casing. Axial motion of the sliding tube is obtained by screwed rods and nuts connected by chain gearing to a handwheel fixed near the control panel. The shape of the dished flanges is designed to convert the kinetic energy of the air leaving the impeller into pressure energy in the casing, from which the air issues at high velocity upon the engine. In its passage through the casing and discharge opening the air passes through wind straighteners which correct any tendency to whirl such as might interfere with the true reading of the Torque. The impeller can be entirely masked by the sliding tube, and in this state the resistance to rotation is a minimum; but by withdrawing the tube the resistance can be made sufficiently large to absorb the entire power of the engine over a range of speed.

S. A. E. Engine Testing Procedure

Engine testing forms for recording engine tests consist of four sheets: A—Rules and Directions; B—Specification Sheet; C—Log Sheet; D—Curve Sheet. Engine testing forms, printed on 8½ by 11-inch punched

sheets, may be obtained in any quantity from the office of the Society.

RULES AND DIRECTIONS—A

A complete engine test includes the determination at different speeds of: maximum horsepower; fuel economy at maximum, at ¾, at ½ and at ¼ maximum horsepower at each of the speeds; friction-horsepower. From these data the following curves are plotted on the Curve Sheet:

(1) Torque against revolutions per minute.
(2) Maximum horsepower against revolutions per minute.
(3) Brake mean effective pressure against revolutions per minute.
(4) Friction horsepower against revolutions per minute.
(5) Mechanical efficiency against revolutions per minute.
(6) Fuel per brake horsepower hour against maximum horsepower at each speed.
(7) Fuel per brake horsepower hour against ¾ maximum horsepower at each speed.

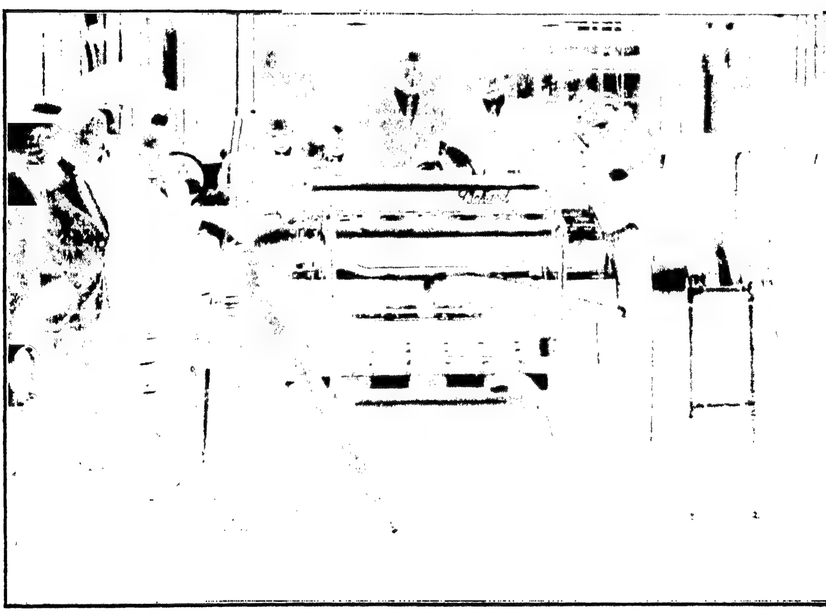


Fig. 52.—Photograph Showing Packard 24-Cylinder X Model, Coupled to Dynamometer Being Demonstrated for the German Transoceanic Fliers, Baron Von Huenfeld, Captain Koehl at the Throttle, and Major Fitzmaurice of the Irish Free State Air Corps.

(8) Fuel per brake horsepower hour against ½ maximum horsepower at each speed.
(9) Fuel per brake horsepower hour against ¼ maximum horsepower at each speed.
(10) Thermal efficiency against maximum horsepower at each speed.
(11) Thermal efficiency against ¾ maximum horsepower at each speed.
(12) Thermal efficiency against ½ maximum horsepower at each speed.
(13) Thermal efficiency against ¼ maximum horsepower at each speed.

Emphasis is laid upon the value of the determination, in addition to the usual runs at maximum horsepower at each speed, of fuel consumption and thermal efficiency at each speed with the engine developing ¾, ½ and ¼ of its maximum horsepower *at that speed.* Automobile engines operate a large proportion of the time on part load and in the study of operating characteristics, part horsepower curves are valuable. At the other hand, aviation engines operate at from ¾ to full load and performances at speeds giving from ¾ to full horsepower are most valuable in studying the characteristics of such engines.

During the complete test, the engine shall be controlled by means of

S.A.E. ENGINE TESTING FORMS **LOG SHEET—C No.______**

Name______________________ Fuel______B T U per Lb______ Sp Grav______at______deg fahr.
Model______________________No Cyls.______ Dynamometer______________________Arm (R)______ft
Bore______In. Stroke______In Displ. (D)______Cu In. Humidity______________per cent.
Laboratory______________________Date______ Oil______________Grade______________Cold Test______deg fahr
Observers______________________ Saybolt Univ Vis at 100 deg fahr______At 210 deg fahr______

	RUN NUMBER	SYMBOL	FORMULA	1	2	3	4	5	6	7	8	9	10
BRAKE HORSEPOWER AND FUEL CONSUMPTION	TIME STARTED		*										
	DURATION OF RUN—MIN	t	*										
	COUNTER START	C_0	*										
	COUNTER FINISH	C_1	*										
	TOTAL REV	r	$C_1 - C_0$										
	AVERAGE R P M	N	$\frac{r}{t}$										
	BRAKE LOAD AT ARM R	P	*										
	TORQUE LB FT OBSERVED	T	PR										
	TORQUE CORRECTED												
	BRAKE M E P	b_p	$\frac{150.8\ T}{D}$										
	BRAKE HP OBSERVED	B HP	$\frac{PRN}{5252\ T}$										
	BRAKE HP CORRECTED												
	FRICTION HP AT N	F HP	F HP Curve										
	INDICATED HP OBSERVED	I HP	B HP + F HP										
	INDICATED HP CORRECTED												
	MECHANICAL EFFICIENCY	M E	$\frac{B\ HP}{I\ HP}$										
	TEMP JACKET WATER—IN		*										
	TEMP JACKET WATER—OUT		*										
	TEMP OIL—IN		*										
	TEMP OIL—OUT		*										
	OIL PR LBS		*										
	TEMP AIR TO CARBURETER		*										
	ROOM TEMP		*										
	WT FUEL START	W_0	*										
	WT FUEL FINISH	W_1	*										
	LB FUEL USED	W	$W_0 - W_1$										
	LB FUEL PER B HP HR	F	$\frac{60\ W}{t \times B\ HP}$										
	THERMAL EFF RE B HP	T E	$\frac{2545}{F \times B\ T\ U}$										
	BAROMETER IN MERCURY												
FRICTION HP.	TIME STARTED		*										
	DURATION OF RUN—MIN	t	*										
	COUNTER START	C_0	*										
	COUNTER FINISH	C_1	*										
	AVERAGE R P M	n	$\frac{C_1 - C_0}{t}$										
	BRAKE LOAD AT ARM R	p	*										
	FRICTION HP	F HP	$\frac{pRn}{5252\ T}$										
	MEAN TEMP JACKET WATER		*										

* Laboratory readings.

Refer to Specification Sheet No.______

Refer to Curve Sheet No.______

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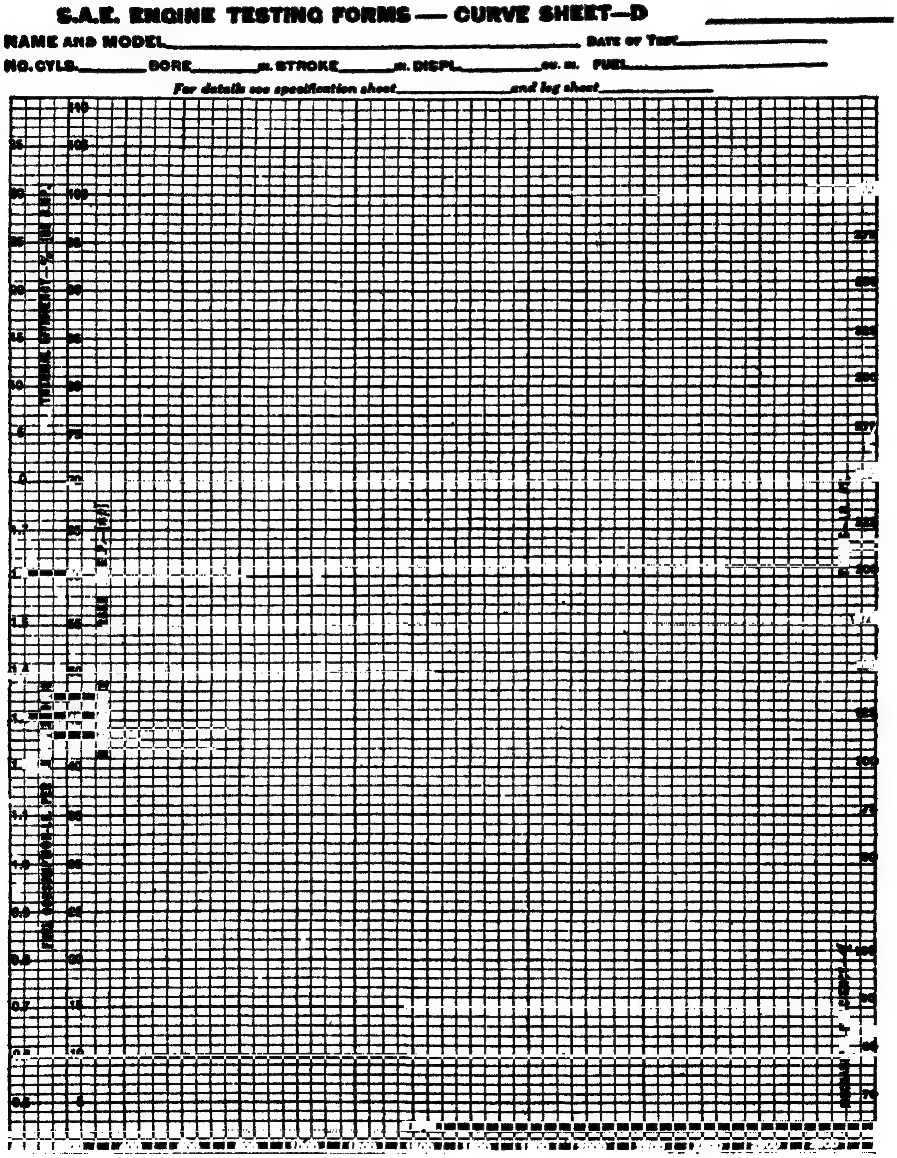
throttle and spark only. Engine adjustments shall be made for best horsepower output, and in no case shall such adjustments be changed during the complete test.

Test runs should not be made until the engine has been run-in sufficiently to show no appreciable difference in friction before and after a run of 30 minutes at the speed of maximum horsepower with the throttle wide open.

Where a stock engine is to be tested, all parts, accessories and lubricants must be stock. In every case, all regular equipment, such as the fan



and generator, must be on the engine and operating.

Before beginning any run, the engine should be brought to a condition of sustained operation under the conditions of the run and it is imperative that in every case the revolutions per minute, brake loads, rate of fuel consumption, cooling-water temperatures, oil temperatures, air draft and other factors remain substantially constant, steady and sustained throughout the run. Flash readings and tests are unscientific and misleading.

Number of Runs.—In every test, enough runs should be taken throughout the speed range so that the points therefor when plotted will indicate clearly the shape and characteristics of the curves. For horsepower and fuel economy tests, it is recommended that runs be made at intervals of approximately 200 r.p.m. A run should be made at the lowest steady operating speed of the engine. All points from which curves are plotted are to be clearly shown on the Curve Sheet.

Duration of Runs.—The duration of brake-horsepower tests shall not be less than three minutes. Where fuel consumption is measured, the duration of tests shall not be less than five minutes. The duration of friction-horsepower tests shall not be less than one minute. The above stated times are minima. In most instances it is desirable to make the runs of longer duration, and it is imperative that in every case revolutions per minute, brake loads, rate of fuel consumption, cooling-water temperatures, oil temperature, air draft and other factors remain substantially constant and steady throughout the run.

Balancing Dynamometer.—Before brake load readings are recorded, great care should be exercised to see that the dynamometer itself is properly balanced. For the electric-cradle type of dynamometer, balancing is accomplished as follows:—The dynamometer is run idle as a motor (drawing current from the line) and a suitable counterbalance on the field frame (which should be perfectly free to turn within limits in ball bearing trunnions) is then adjusted so that the platform scales read zero. This reading should be obtained with the dynamometer rotating first in one direction and then in the other. The reaction of the armature on the field frame will exactly balance the friction of the brushes and armature bearings carried in the field frame. With the armature still rotating, check-weights (or pieces of metal having a known weight) should be hung from the knife-edge on the dynamometer arm. If the reading recorded by the platform scales is equal to the known weight applied, the dynamometer can be considered as balanced.

Brake Loads.—Brake load readings should be taken with accurately calibrated platform or beam scales. The connection of the dynamometer arm to these scales by means of knife-edges, calibrated spring balance and tripod or suitable linkage is recommended. Suitable counter-balances or tare loads must be accurately provided. The spring balance gives a quick approximate reading for brake loads as it serves to cushion the platform or beam scales from shock and vibration. During any run, the platform or beam scales should be kept balanced, and the loads registered thereby must be substantially constant and steady throughout the run.

Revolutions per Minute.—Speed in revolutions per minute should be invariably taken from positively driven counters which engage at the be-

ginning of the run and disengage at the end. The difference between the two readings, divided by the duration of run in minutes, gives the true average speed. Tachometers, even though carefully calibrated, are not sufficiently reliable. In connection with the speed counters mentioned however the tachometer may be used as an approximate check on average speed, also as an indicator of variations in speed before or during the run.

The maximum allowable variation in speed during a run shall be 50 r.p.m.

Temperatures.—All temperatures are to be given in degrees Fahrenheit.

A reliable glass straight-stem thermometer should be placed near the carburetor air-inlet in order to measure the temperature of the entering air. This thermometer should be read at least three times during each run, one of these times to be at beginning and one at end of run.

Thermometers should also be placed in suitable wells or sockets, one near the inlet of the pump and another as close as possible to the water-outlet of the engine. These wells or sockets should be in pipes that run full, so that water continually circulates about them. They should be filled with oil or mercury, and careful readings taken at least three times during each run, one of these times to be at beginning and one at end of run.

In order to afford a fixed basis of comparison, it is recommended that the outlet water temperature for engines be kept at 175 degrees Fahrenheit plus or minus five degrees. Control of the outlet temperature can be accomplished by thermostat located in the outlet line or by external control of quantity or temperature of inlet water. Where the thermostat or other cooling water regulating devices are stock, these may be attached and operating during a test.

In every case, inlet and outlet cooling-water temperatures should remain substantially constant and steady throughout a run. The maximum allowable variation in cooling-water temperature shall be ten degrees Fahrenheit.

During friction-horsepower runs it is desired to obtain the mean temperature of the jacket water. If the water is pump-circulated, the average of the inlet outlet temperatures may be taken. If thermosyphon circulation is used, the water will not circulate noticeably during a friction-horsepower run. The mean jacket-water temperature for such engines can be taken by inserting thermometers into the jacket space, the average of readings being taken. In every case of friction-horsepower test, the test must be made immediately after the corresponding brake-horsepower test, before the engine has cooled.

An air draft should be provided which approximates in amount and effect the air draft on the road with the car moving at a speed corresponding to the given engine speed. During friction-horsepower tests, of course, this air draft is shut off, in order not to cool the engine.

For air-cooled engines, the air draft is of the greatest importance.

Friction-Horsepower.—The approximate friction-horsepower of an engine can be measured best by means of an electric dynamometer, preferably of the cradle type. The dynamometer is used to drive the engine under test at various speeds, and the torque reaction is measured. This will be in the opposite direction to that obtaining while the engine is driving the dynamometer, so that provision must be made for measuring the torque on

both sides of the dynamometer, or else suitable linkage must be provided to change the direction of the pull. The test for friction-horsepower should be made immediately after the brake-horsepower test, before the engine has cooled, in order to keep the condition of the lubricating oil and the friction of the parts the same as during the brake-horsepower test, as nearly as possible. During this test the throttle of the engine should remain in the same position as for the corresponding brake-horsepower test. Compression-relief cocks should remain closed and all accessories, such as magneto, generator and pumps, used during the brake-horsepower test, should be in operation.

Indicated-Horsepower.—Approximate indicated-horsepower is obtained by adding to the brake-horsepower at any given speed the friction-horsepower obtained at the same speed.

If the friction-horsepower and brake-horsepower tests are not made at exactly the same speeds, the friction-horsepower at any given speed can be obtained from the friction-horsepower curve plotted on the Curve Sheet. Tedious interpolation is thus avoided.

If it is desired to correct the results, a standard barometric pressure of 29.92 inches of mercury and a standard temperature of 500 degrees Absolute, which is the same as 60 degrees Fahrenheit, shall be used in making the corrections.

Correction Formula

$$\text{B-Hp}_c = \text{B-Hp}_o \times \frac{P_s}{P_o} \times \sqrt{\frac{T_o}{T_s}}$$

where

B-Hp_c = corrected brake-horsepower

B-Hp_o = observed brake-horsepower

P_o = observed barometric pressure in inches of mercury

P_s = standard barometric pressure of 29.92 inches of mercury

T_o = observed absolute temperature in degrees Fahrenheit

T_s = standard absolute temperature of 520 degrees Fahrenheit

Rules and Directions for Use of Forms

Specification Sheet.—(3) The compression volume is the volume occupied by the charge when the piston is at the top of the compression stroke. To measure this volume with the piston on dead center at end of compression stroke (with both valves closed) fill the compression space from a graduate containing a known volume of light oil. Care must be taken to correct for leakage. The total volume of the cylinder equals the piston displacement plus the compression volume. Give compression pressure at speed of maximum torque, or at speed of standard starter.

(4) State number of cylinders cast integral, whether offset, type of cylinder head, whether water space is provided between adjacent cylinders.

(6) State whether water or air-cooled. If the former, state whether pump or thermosyphon. Note if two pumps or thermostat are used. State type of pump. Give the diameter of the fan, the number and projected width of the fan blades and the ratio of the fan speed to the engine speed.

(7) Weight of piston with rings and pin should include weight of bush-

SPECIFICATION SHEET—B.

Name and Model ....Date of Test ....
Manufacturer ....
(1)[1] General Type ....Cycle ....
(2) No. of Cyl....Bore...in., Stroke...in., Piston Displ. per Cyl....cu. in., Tot....Cu. in.
(3) Compression Vol. (V_c) ....cu. in., Total Vol. of Cyl. (V)...........cu. in.,

$$\text{Compression Ratio} = \frac{\text{Comp. Vol.}}{\text{Total Vol.}} = \frac{V_c}{V} = \text{..Comp. Pressure... lb. gauge at..r.p.m.}$$

(4) Type of Cyl. Casting ....Matl. ....
(5) Type of Valves ....Location ....
(6) Cooling System ....
Fan Diam...in. No. of Blades..Projected Width..in. Ratio of Fan to Engine Speed..
(7) Piston, Type .... Matl.....................
Wt. with Rings and Pin..lb., Length..in., Distance Center of Pin to Top of Piston..in.
(8) Piston-Rings, No. per Piston ......Type ....Width ....in.
(9) Connecting-Rod, Type ....
Length, c. to c. ....in. Weight, Upper End ....lb., Lower End ....lb., Total ....lb.
(10) Piston-Rod Bearings, Diam. ....in., Total Length ....in., Matl. ....Location........
(11) Connecting-Rod Bearings, Diam. ....in., Length .... in., Matl. .... Type...........
(12) Crankshaft Bearings, No. ....Diams. ....Material ....Lengths ....
(13) Camshaft Bearings, No. ....Diams. ....Material ....Lengths ....
(14) Type of Cams ....Type of Valve-Lifters ....
(15) Inlet Valves. No. per Cyl.....o.d....in., Port Diam....in., Lift..in., Seat Angle..deg.
(16) Exhaust Valves. No. per Cyl...o.d....in., Port Diam....in., Lift..in., Seat Angle..deg.
(17) Weight of Valve Reciprocating Parts, Inlet ....lb., Exhaust ....lb.
(18) Valve-Spring Tension, Inlet Open...lb., Closed...lb., Exhaust Open...lb., Closed..lb.
(19) Valve-Timing, Inlet Valve Opens...deg...Top Center, Closes..deg. after Low. Center
Exhaust Valve Opens...deg. before Lower Center, Closes...deg....Top Center
(20) Flywheel, o.d. ......in., Weight ...... lb. Moment of Inertia ....
(21) Weight of Engine ....lb., Including ....

Carburetion

(22) Carburetor, Name and Model ....Nom. Size ....in.
(23) Specifications (Size of Nozzles, etc.) ....
(24) How Heated ....
(25) General Principles of Operation ....
....
(26) Description of Intake Pipe ....
....

Ignition

(27) Name and Type of System ....
(28) Type of Distributor ....Firing Order.......................
(29) Type of Breaker ....Maximum Spark Advance ......deg., Retard ......deg.
(30) Sparkplugs, Name and Type ....Size ....in.
(31) Location ....Gap ....in.

Lubrication System

(32) Type and Description ....
....

Accessories

(33) Accessories Attached During Test ....
....

ings, screws, or other piston-pin fastening devices in the piston. Record all weights in pounds and decimal parts thereof. In measuring length of piston and distance from center of pin to top of piston, deduct any chamfer or crown at top of piston.

(8) Specify whether rings are concentric or eccentric; give name, sketch or description of special types. If oil-ring is used, state location.

In fact, it is necessary to fill out the specification sheet which follows in detail. The data called for by items nine to 21 inclusive are obtained from the working drawings of the engine for the most part. Items 22 and 23 are obtained from the carburetor manufacturer.

(24) State if heated by water or exhaust, and whether part or all of the air entering carburetor is heated.

(25) Under "general principles of operation" give description. (Venturi type with single adjustable nozzle and single auxiliary air-valve with one spring and straight-tube type, four non-adjustable nozzles coming into operation successively as air-flow increases.)

(26) By description and sketch, give general form, approximate inside diameters of different portions, and specify which, if any, parts are jacketed.

(27) In case of two systems, state which was used in test.

(29) State if spark is fixed, or if spark control is automatic or manual. Maximum spark advance and retard are to be given in degrees of crankshaft rotation.

(30) Give material of insulation, number of sparking points on electrodes.

(31) In addition to exact location in combustion-chamber, state whether vertical, horizontal or inclined, and whether plug extends into combustion-chamber.

(32) Give the general type of lubrication system (recirculating splash; force-feed and spray and complete force-feed). Then describe in detail action of system and course taken by oil. State oil pressures and type of pump used.

(33) Give list and description of accessories attached during the test.

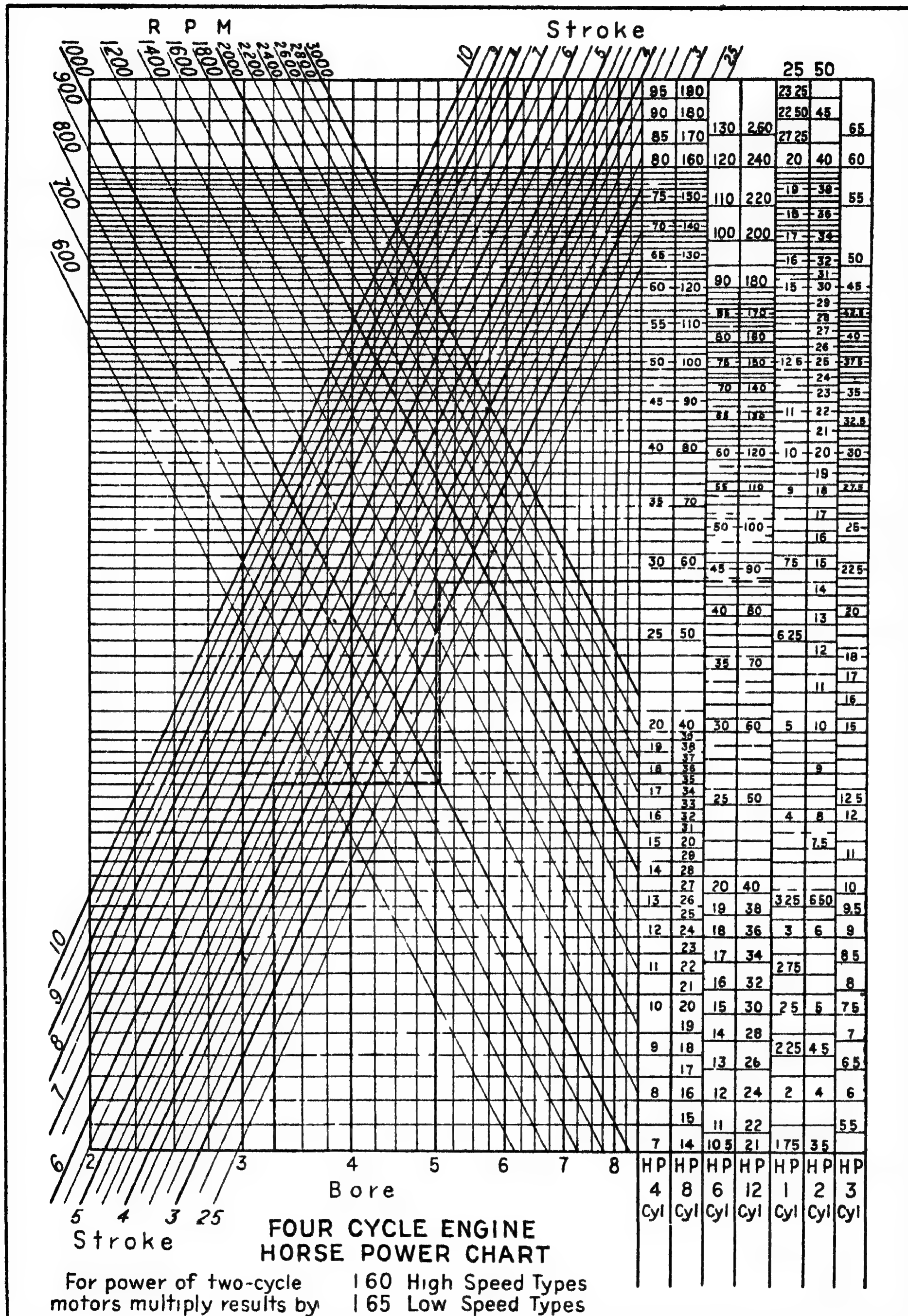
QUESTIONS FOR REVIEW

1. What is the value of the indicator?
2. Why cannot the same indicator be used on steam engines and high-speed gas engines?
3. What is the principle of the fan brake; of the water brake?
4. Why do air-cooled engines require special testing apparatus?
5. What is friction horsepower and how is it determined?

Circumferences and Areas of Circles

Diameters from $\frac{1}{64}$ inch to $9\frac{7}{8}$ inches

Diam.	Circum.	Area	Diam.	Circum.	Area	Diam.	Circum.	Area
1/64	.04909	.00019	2	6.2832	3.1416	5	15.708	19.635
1/32	.09818	.00077	1/16	6.4795	3.3410	1/16	15.904	20.129
3/64	.14726	.00173	1/8	6.6759	3.5466	1/8	16.101	20.629
1/16	.19635	.00307	3/16	6.8722	3.7583	3/16	16.297	21.135
3/32	.29452	.00690	1/4	7.0686	3.9761	1/4	16.493	21.648
1/8	.39270	.01227	5/16	7.2649	4.2000	5/16	16.690	22.166
5/32	.49087	.01917	3/8	7.4613	4.4301	3/8	16.886	22.691
3/16	.58905	.02761	7/16	7.6576	4.6664	7/16	17.082	23.221
7/32	.68722	.03758	1/2	7.8540	4.9087	1/2	17.279	23.758
1/4	.78540	.04909	9/16	8.0503	5.1572	9/16	17.475	24.301
9/32	.88357	.06213	5/8	8.2467	5.4119	5/8	17.671	24.850
5/16	.98175	.07670	11/16	8.4430	5.6727	11/16	17.868	25.406
11/32	1.0799	.09281	3/4	8.6394	5.9396	3/4	18.064	25.967
3/8	1.1781	.11045	13/16	8.8357	6.2126	13/16	18.261	26.535
13/32	1.2763	.12962	7/8	9.0321	6.4918	7/8	18.457	27.109
7/16	1.3744	.15033	15/16	9.2284	6.7771	15/16	18.653	27.688
15/32	1.4726	.17257	3	9.4284	7.0686	6	18.850	28.274
1/2	1.5708	.19635	1/16	9.6211	7.3662	1/8	19.242	29.465
17/32	1.6690	.22166	1/8	9.8175	7.6699	1/4	19.635	30.680
9/16	1.7671	.24850	3/16	10.014	7.9798	3/8	20.028	31.919
19/32	1.8653	.27688	1/4	10.210	8.2958	1/2	20.420	33.183
5/8	1.9635	.30680	5/16	10.407	8.6179	5/8	20.813	34.472
21/32	2.0617	.33824	3/8	10.603	8.9462	3/4	21.206	35.785
11/16	2.1598	.37122	7/16	10.799	9.2806	7/8	21.598	37.122
23/32	2.2580	.40574	1/2	10.996	9.6211	7	21.991	38.485
3/4	2.3562	.44179	9/16	11.192	9.9678	1/8	22.384	39.871
25/32	2.4544	.47937	5/8	11.388	10.321	1/4	22.776	41.282
13/16	2.5525	.51849	11/16	11.585	10.680	3/8	23.169	42.718
27/32	2.6507	.55914	3/4	11.781	11.045	1/2	23.562	44.179
7/8	2.7489	.60132	13/16	11.977	11.416	5/8	23.955	45.664
29/32	2.8471	.64504	7/8	12.174	11.793	3/4	23.347	47.173
15/16	2.9452	.69029	15/16	12.370	12.177	7/8	24.740	48.707
31/32	3.0434	.73708	4	12.566	12.566	8	25.133	50.265
1	3.1416	.7854	1/16	12.763	12.962	1/8	25.525	51.849
1/16	3.3379	.8866	1/8	12.959	13.364	1/4	25.918	53.456
1/8	3.5343	.9940	3/16	13.155	13.772	3/8	26.311	55.088
3/16	3.7306	1.1075						
1/4	3.9270	1.2272	1/4	13.352	14.186	1/2	26.704	56.745
5/16	4.1233	1.3530	5/16	13.548	14.607	5/8	27.096	58.426
3/8	4.3197	1.4849	3/8	13.744	15.033	3/4	27.489	60.132
7/16	4.5160	1.6230	7/16	13.941	15.466	7/8	27.882	61.862
1/2	4.7124	1.7671	1/2	14.137	15.904	9	28.274	63.617
9/16	4.9087	1.9175	9/16	14.334	16.349	1/8	28.667	65.397
5/8	5.1051	2.0739	5/8	14.530	16.800	1/4	29.060	67.201
11/16	5.3014	2.2365	11/16	14.726	17.257	3/8	29.452	69.029
3/4	5.4978	2.4053	3/4	14.923	17.728	1/2	29.845	70.882
13/16	5.6941	2.5802	13/16	15.119	18.190	5/8	30.238	72.760
7/8	5.8905	2.7612	7/8	15.315	18.665	3/4	30.631	74.662
15/16	6.0868	2.9483	15/16	15.512	19.147	7/8	31.023	76.589



Directions for use—Run vertical line from bore to diagonal representing stroke, then horizontally to right till it intersects r.p.m. diagonal marked 1000 r.p.m.—then vertically to diagonal representing r.p.m. determination is to be made at, then horizontally to right, reading Hp. directly in column showing number of cylinders.

THE METRIC SYSTEM OF MEASUREMENT

Measures of Length

1 Millimeter (mm.)=....................0.03937079 inch, or about 1/25 inch

10 Millimeters=1 Centimeter (cm.)=....................0.3937079 inch

10 Centimeters=1 Decimeter (dm.)=....................3.937079 inch

10 Decimeters=1 meter (m.)=..........39.37079 inches, 3.2808992 feet, or 1.09361 yards

10 Meters=1 Decameter (Dm.)=....................32.808992 feet

10 Decameters=1 Hectometer (Hm.)=....................19.927817 rods

10 Hectometers=1 Kilometer (Km.)=..........1093.61 yards, or 0.6213824 mile

10 Kilometers=1 Myriameter (Mm.)=....................6.213824 miles

1 inch=2.54 cm., 1 foot=0.3048 m., 1 yard=0.9144 m., 1 rod=0.5029 Dm., 1 mile=1.6093 Km.

Measures of Weight

1 Gram (g.)=..........15.4324874 gr. Troy, or 0.03215 oz. Troy, or 0.03527398 oz. avoir.

10 Grams=1 Decagram (Dg.)=....................0.3527398 oz. avoir.

10 Decagrams=1 Hectogram (Hg.)=....................3.527398 oz. avoir.

10 Hectograms=1Kilogram (Kg.)=....................2.20462125 lbs.

1000 Kilograms=1 Ton (T.)=2204.62125 lbs., or 1.1023 tons of 2000 lbs., or 0.9842 ton of 2240 lbs., or 19.68 cwts.

1 grain=0.0648 g., 1 oz. avoir.=28.35 g., 1 lb.=0.4536 Kg., 1 ton 2000 lbs.=0.9072 T., 1 ton 2240 lbs.=1.016 T., or 1016 Kg.

Measures of Capacity

1 Liter (l.)=1 cubic decimeter=61.0270515 cubic in., or 0.03531 cu. ft., or 1.0567 liquid qts., or 0.908 dry qt., or 0.26417 Amer. gal.

10 Liters=1 Decaliter (Dl.)=2.6417 gal., or 1.135 pk.

10 Decaliters=1 Hectoliter (Hl.)=2.8375 bu.

10 Hectoliters=1 Kiloliter (Kl.)=61027.0515 cu. in., or 28.375 bu.

1 cu. foot=28.317 l., 1 gallon, Amer.=3.785 l., 1 gallon Brit.=4.543 l.

A CONVENIENT METRIC CONVERSION

Compiled by C. W. Hunt

Millimeters×.03937=inches.
Millimeters÷25.4=inches.
Centimeters×.3937=inches.
Centimeters÷2.54=inches.
Meters×39.37=inches. (Act of Congress.)
Meters×3.281=feet.
Meters×1.094=yards.
Kilometers×.6214=miles.
Kilometers÷1.6093=miles.
Kilometers×3280.8=feet.
Square Millimeters×.00155=square inches.
Square Millimeters÷645.2=square inches.
Square Centimeters×.155=square inches.
Square Centimeters÷6.452=square inches.
Square Meters×10.764=square feet.
Square Kilometers×247.1=acres.
Hectare×2.471=acres.
Cubic Centimeters÷16.387=cubic inches.
Cubic Centimeters÷3.697=fluid drams.
Cubic Centimeters÷29.57=fluid ounces.
Cubic Meters×35.314=cubic feet.
Cubic Meters×1.308=cubic yards.
Cubic Meters×264.2=gallons (231 cubic inches).
Liters×61.023=cubic inches. (Act of Congress.)
Liters×33.84=fluid ounces.
Liters×.2642=gallons (231 cubic inches).
Liters÷3.785=gallons (31 cubic inches).
Liters÷28.317=cubic feet.
Hectoliters×3.531=cubic feet.
Hectoliters×2.838=bushels (2150.42 cubic inches).
Hectoliters×.1308=cubic yards.
Hectoliters×26.42=gallons (231 cubic in.).
Grams×15.432=grains. (Act of Congress.)
Grams×981=dynes.
Grams÷28.35=ounces avoirdupois.
Juole×.7373=foot pounds.
Kilograms×2.2046=pounds.
Kilograms×35.27=ounces avoirdupois.
Kilograms÷907.2=tons (2,000 pounds).
Kilograms per square cent.×14.223=pounds per square inch.
Kilogrammeters×7.233=foot lbs.
Kilo per Meter×.672=lbs. per foot.
Kilo per Cubic Meter×.0624=lbs. per cubic foot.
Kilowatts×1.34=horsepower.
Watts÷746=horsepower.
Watts×.7373=ft. pounds per second.
Caloric×3.968=B. T. U.
Cheval vapeur×.9863=horsepower.
Centigrade×1.8+32=degree Fahrenheit.
Gravity Paris=980.94 Centimeter per sec.

UNITED STATES OFFICIAL MILLIMETER CONVERSION TABLE

Millimeters	Equivalent in Inches	Millimeters	Equivalent in Inches
1	0.03937	51	2.00787
2	0.07874	52	2.04724
3	0.11811	53	2.08661
4	0.15748	54	2.12598
5	0.19685	55	2.16535
6	0.23622	56	2.20472
7	0.27559	57	2.24409
8	0.31496	58	2.28346
9	0.35433	59	2.32283
10	0.39370	60	2.36220
11	0.43307	61	2.40157
12	0.47244	62	2.44094
13	0.51181	63	2.48031
14	0.55118	64	2.51968
15	0.59055	65	2.55905
16	0.62992	66	2.59842
17	0.66929	67	2.63779
18	0.70866	68	2.67716
19	0.74803	69	2.71653
20	0.78740	70	2.75590
21	0.82677	71	2.79527
22	0.86614	72	2.83464
23	0.90551	73	2.87401
24	0.94488	74	2.91338
25	0.98425	75	2.95275
26	1.02362	76	2.99212
27	1.06299	77	2.03149
28	1.10236	78	3.07086
29	1.14173	79	3.11023
30	1.18110	80	3.14960
31	1.22047	81	3.18897
32	1.25984	82	3.22834
33	1.29921	83	3.26771
34	1.33858	84	3.30708
35	1.37795	85	3.34645
36	1.41732	86	3.38582
37	1.45669	87	3.42519
38	1.49606	88	3 46456
39	1.53543	89	3.50393
40	1.57480	90	3.54330
41	1.61417	91	3.58267
42	1.65354	92	3.62204
43	1.69291	93	3.66141
44	1.73228	94	3.70078
45	1.77165	95	3.74015
46	1.81102	96	3.77952
47	1.85039	97	3.81889
48	1.88976	98	3.85826
49	1.92913	99	3.89763
50	1.96850	100	3.93700

DECIMAL FRACTIONS OF A LINEAR INCH IN MILLIMETERS

Inch	Millimeters	Inch	Millimeters	Inch	Millimeters	Inch	Millimeters
.01	.254	.29	7.366	.57	14.478	.85	21.590
.02	.508	.30	7.620	.58	14.732	.86	21.844
.03	.762	.31	7.874	.59	14.986	.87	22.098
.04	1.016	.32	8.128	.60	15.240	.88	22.352
.05	1.270	.33	8.382	.61	15.494	89	22.606
.06	1.524	.34	8.636	.62	15.748	.90	22.860
.07	1.778	.35	8.890	.63	16.002	.91	23.114
.08	2.032	.36	9.114	.64	16.256	.92	23.368
.09	2.286	.37	9 398	.65	16.510	.93	23.622
.10	2 540	.38	9.652	.66	16.764	.94	23.876
.11	2.794	39	9.906	.67	17 018	.95	24.130
.12	3.048	.40	10.160	.68	17.272	.96	24.384
.13	3.302	.41	10.414	.69	17.526	.97	24.638
.14	3.556	.42	10.668	.70	17.780	98	24.892
.15	3.810	.43	10.922	.71	18.034	.99	25.146
.16	4.064	.44	11.176	.72	18.288	1.00	25.400
.17	4.318	.45	11.430	.73	18.542	2.00	50.799
.18	4.572	.46	11.684	.74	18.796	3.00	76.199
.19	4.826	.47	11.938	.75	19.050	4.00	101.598
.20	5.080	.48	12.192	.76	19.304	5.00	126 998
.21	5.334	.49	12.446	.77	19.558	6.00	152.397
.22	5.588	.50	12 700	.78	19.812	7.00	177 797
.23	5.842	.51	12.945	.79	20 066	8.00	203.196
.24	6.096	.52	13.208	.80	20.320	9.00	228.596
.25	6.350	.53	13.462	.81	20.574	10.00	253.995
.26	6.604	.54	13.716	.82	20.828	11.00	279.395
.27	6.858	.55	13.970	.83	21.082	12.00	304.794
.28	7.112	.56	14.224	.84	21.336	1 foot	

DECIMAL EQUIVALENTS OF SIXTY-FOURTHS

$\frac{1}{64}$=.0156	$\frac{17}{64}$=.2656	$\frac{33}{64}$=.5156	$\frac{49}{64}$= .7656
$\frac{1}{32}$=.0312	$\frac{9}{32}$=.2812	$\frac{17}{32}$=.5312	$\frac{25}{32}$= .7812
$\frac{3}{64}$=.0468	$\frac{19}{64}$=.2968	$\frac{35}{64}$=.5468	$\frac{51}{64}$= .7968
$\frac{1}{16}$=.0625	$\frac{5}{16}$=.3125	$\frac{9}{16}$=.5625	$\frac{13}{16}$= .8125
$\frac{5}{64}$=.0781	$\frac{21}{64}$=.3281	$\frac{37}{64}$=.5781	$\frac{53}{64}$= .8281
$\frac{3}{32}$=.0937	$\frac{11}{32}$=.3437	$\frac{19}{32}$=.5937	$\frac{27}{32}$= .8437
$\frac{7}{64}$=.1093	$\frac{23}{64}$=.3593	$\frac{39}{64}$=.6093	$\frac{55}{64}$= .8593
$\frac{1}{8}$=.1250	$\frac{3}{8}$= 3750	$\frac{5}{8}$=.6250	$\frac{7}{8}$= .8750
$\frac{9}{64}$=.1406	$\frac{25}{64}$=.3906	$\frac{41}{64}$=.6406	$\frac{57}{64}$= .8906
$\frac{5}{32}$=.1562	$\frac{13}{32}$=.4062	$\frac{21}{32}$=.6562	$\frac{29}{32}$= .9062
$\frac{11}{64}$=.1718	$\frac{27}{64}$= 4218	$\frac{43}{64}$=.6718	$\frac{59}{64}$= .9218
$\frac{3}{16}$=.1875	$\frac{7}{16}$= 4375	$\frac{11}{16}$= 6875	$\frac{15}{16}$= .9375
$\frac{13}{64}$=.2031	$\frac{29}{64}$=.4531	$\frac{45}{64}$=.7031	$\frac{61}{64}$= .9531
$\frac{7}{32}$=.2187	$\frac{15}{32}$=.4687	$\frac{23}{32}$=.7187	$\frac{31}{32}$= .9687
$\frac{15}{64}$=.2343	$\frac{31}{64}$=.4843	$\frac{47}{64}$=.7343	$\frac{63}{64}$= .9843
$\frac{1}{4}$=.2500	$\frac{1}{2}$=.5000	$\frac{3}{4}$=.7500	1=1.0000

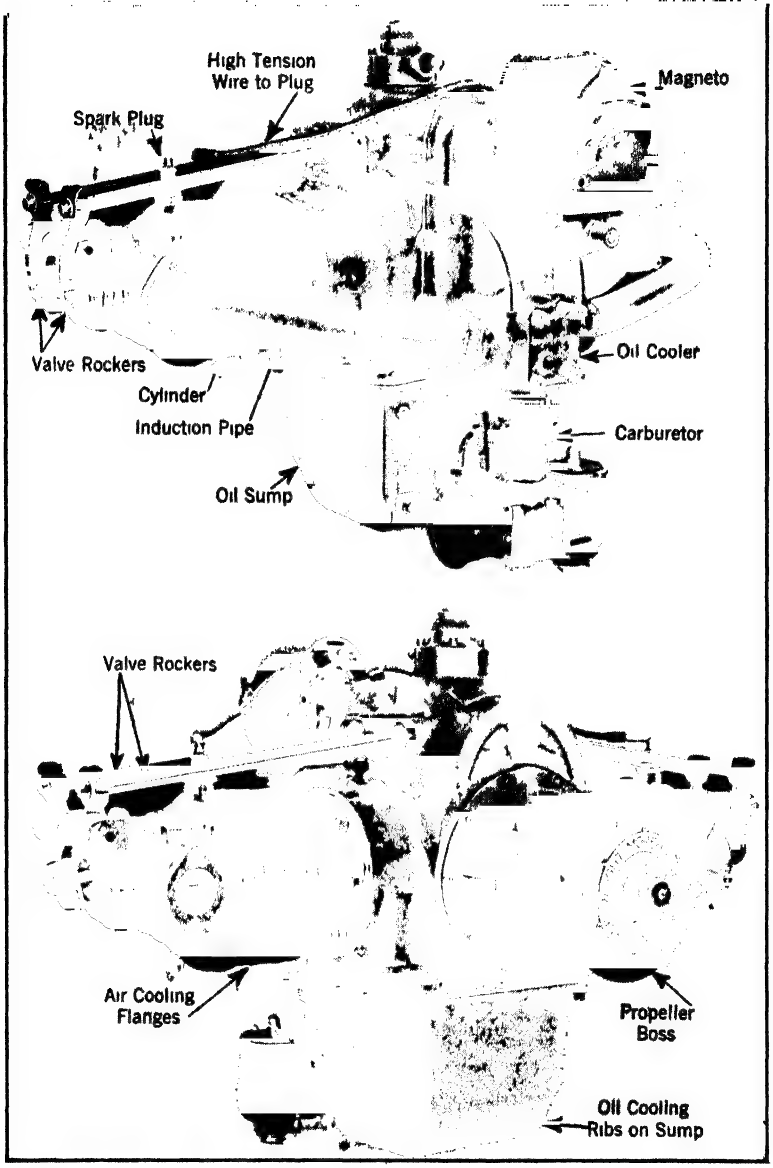


Fig. 53.—The Wright-Morehouse Air-Cooled Engine was Designed for Light Airplanes and was of the Twin Cylinder Opposed Type. View at Top Shows Magneto and Carburetor Placing, that Below Shows Propeller Boss.

mechanical friction loss. Study of the torque outlines or plotted graphics shown at Fig. 70 will indicate how multiplication of cylinders will produce steady power delivery due to overlapping impulses. The most practical form would be that which more nearly conforms to the steady running produced by a steam turbine or electric motor. The advocates of the eight-cylinder engine bring up the item of uniform torque as one of the most important advantages of the eight-cylinder design. A number of torque diagrams are shown at Fig. 70. While these appear to be deeply technical, they may be very easily followed when their purpose is explained. At the top is shown the torque diagram of a single-cylinder motor of the four-cycle type. The high point in the line represents the period of greatest

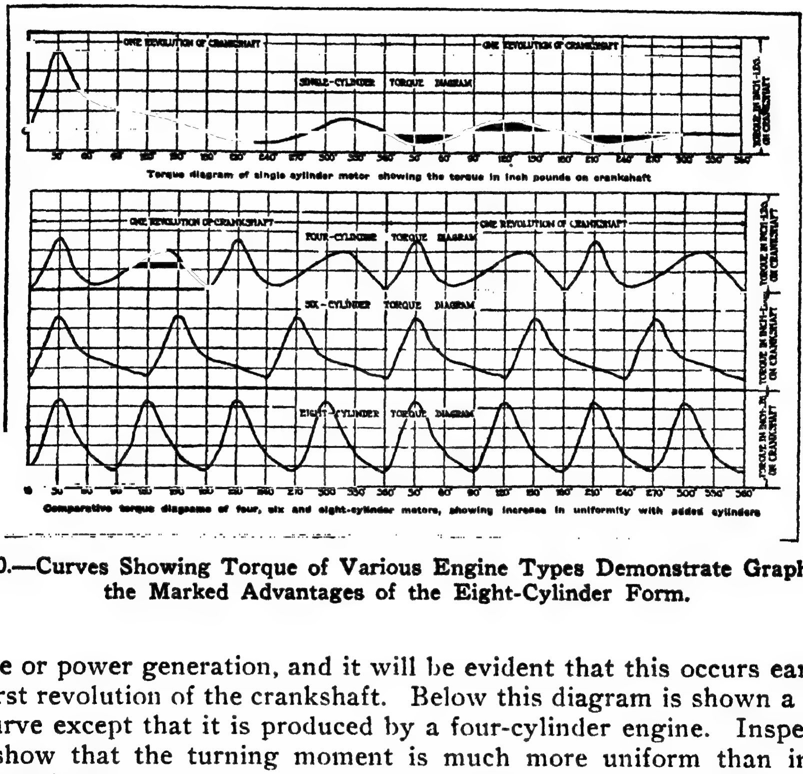


Fig. 70.—Curves Showing Torque of Various Engine Types Demonstrate Graphically the Marked Advantages of the Eight-Cylinder Form.

torque or power generation, and it will be evident that this occurs early in the first revolution of the crankshaft. Below this diagram is shown a similar curve except that it is produced by a four-cylinder engine. Inspection will show that the turning moment is much more uniform than in the single-cylinder; similarly, the six-cylinder diagram is an improvement over the four, and the eight-cylinder diagram is an improvement over the six-cylinder.

The reason that practically continuous torque is obtained in an eight-cylinder engine is that one cylinder fires every 90 degrees of crankshaft rotation, and as each impulse lasts nearly 75 per cent of the stroke, one can easily appreciate that an engine that will give four explosions per revolution of the crankshaft will run more uniformly than one that gives but three explosions per revolution, as the six-cylinder does, and will be twice as effective in promoting smooth running as a four-cylinder, in which but two explosions occur per revolution of the crankshaft. The comparison

regular pattern. Still another method is to have a boss just above the main bearing on one connecting rod to which the lower portion of the connecting rod in the opposite cylinder is hinged. As the eight-cylinder engine may actually be made lighter than the six-cylinder of equal power, it is possible to use smaller reciprocating parts, such as pistons, connecting rods and valve gear, and obtain higher engine speed with practically no vibration. The firing order in nearly every case is the same as in a four-cylinder except

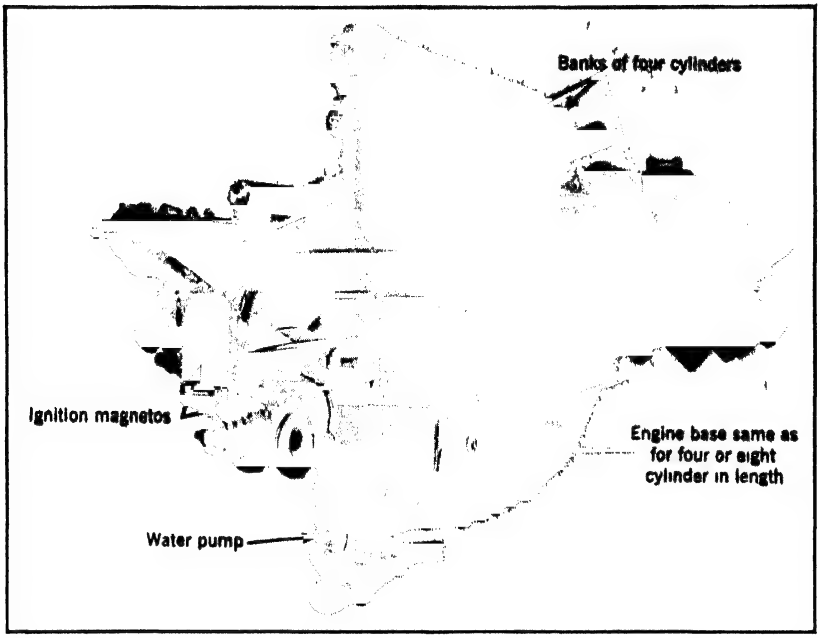


Fig. 74.—Hispano-Suiza Twelve-Cylinder Motor has Three Banks of Four Cylinders Each Arranged in "W" Form. This is Sometimes Called the Broad Arrow Type. The Engine Weighs 390 Kilograms (858 Pounds) and Delivers its Power at 2,000 R.P.M. This View is Taken from the Accessory Drive End.

that the explosions occur alternately in each set of cylinders. The firing order of an eight-cylinder motor is apt to be confusing to the reader, especially if one considers that there are eight possible sequences. The majority of engineers favor the alternate firing from side to side. Firing orders will be considered in proper sequence.

The demand of aircraft designers for more power has stimulated designers to work out twelve-cylinder motors and both the Vee-form and W-types have been used as previously indicated. In the Curtiss Chieftain engine, there is found an unusual arrangement of twelve cylinders. There are six pairs of air-cooled cylinders, one behind the other, spaced equally in static radial relation around the crankcase. These are high-speed motors incorporating all recent features of design in securing light reciprocating parts, large valve openings, etc. The twelve-cylinder motor incorporates the best features of high-speed motor design and there is no need at this time to discuss further the pros and cons of the twelve-cylinder versus the

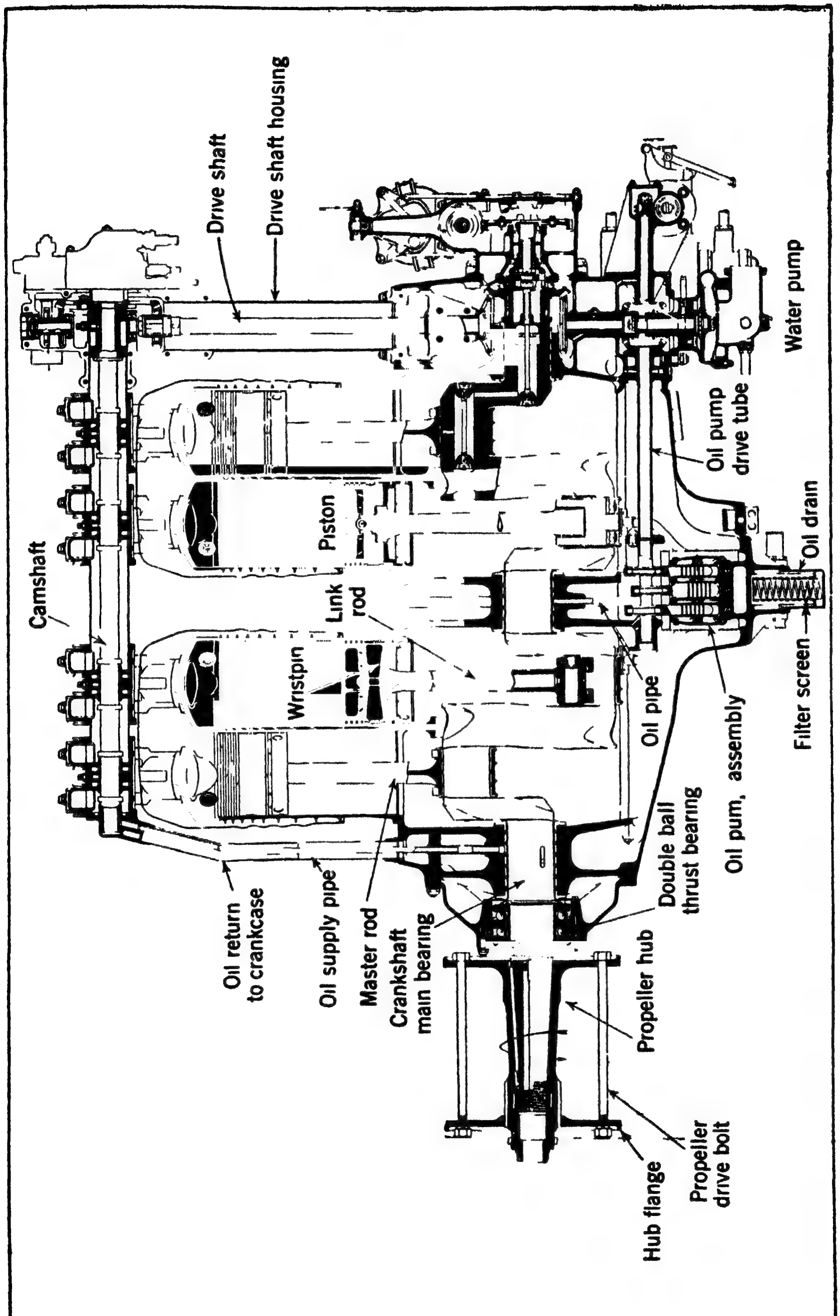


Fig. 75.—Longitudinal Vertical Sectional Drawing of the Lorraine "W" Type Engine Showing Design of Internal Mechanism.

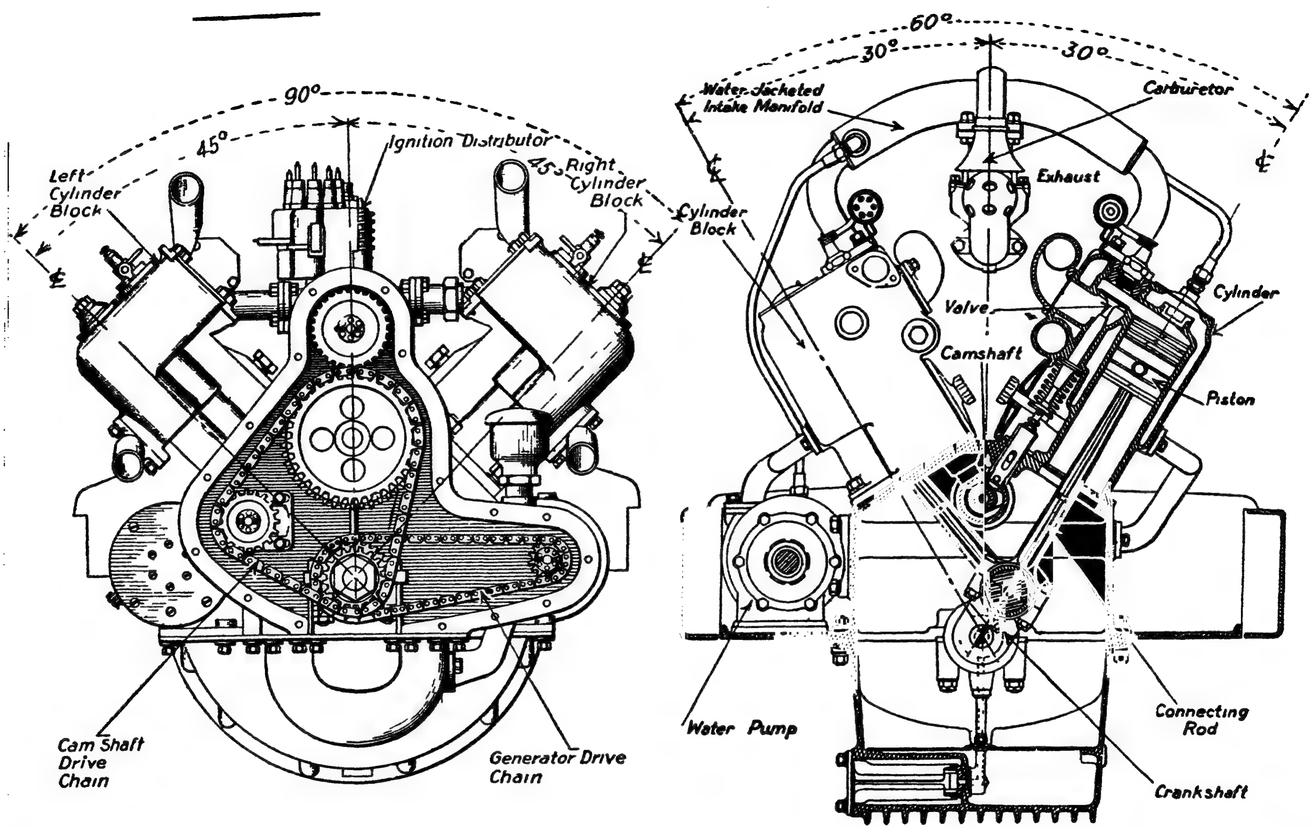


Fig. 76.—How the Angle Between the Cylinders of an Eight- and Twelve-Cylinder "Vee" Motor of the Automobile Type Varies. Some Aviation Engines have Less Than 60 Degrees Included Angle.

eight or six, because it is conceded by all that there is the same degree of steady power application in the twelve over the eight as there would be in the eight over the six. The question resolves itself into having a motor of high power that will run with minimum vibration and that produces smooth action. This is well shown by diagrams previously presented in various forms. It should be remembered that if an eight-cylinder engine will give four explosions per revolution of the flywheel, a twelve-cylinder type will give six explosions per revolution, and instead of the impulses coming 90

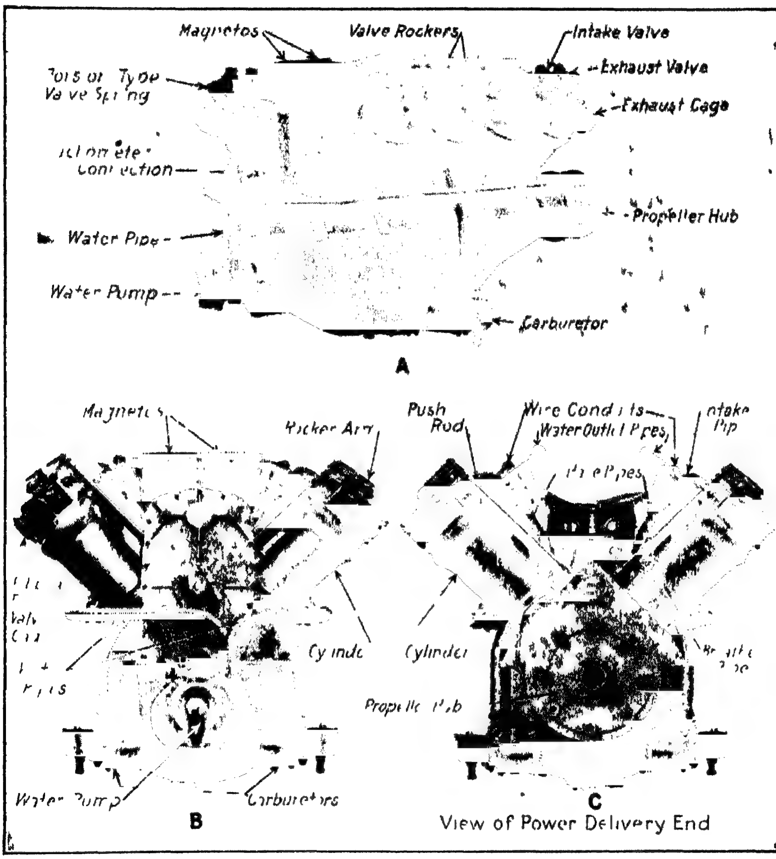


Fig. 77.—The Curtiss Eight-Cylinder 200 Horsepower Aviation Engine, Now an Obsolete Design, Was a Good Example of Pre-War Construction.

degrees crank travel apart, as in the case of the eight-cylinder, these will come but 60 degrees of crank travel apart in the case of the twelve-cylinder. For this reason, the cylinders of a twelve are usually though not always separated by 60 degrees while the eight has the blocks spaced 90 degrees apart if even intervals between explosions are desired. The comparison can be easily made by comparing the sectional views of Vee automotive engines at Fig. 76. When one realizes that the actual duration of the power stroke is considerably greater than 120 degrees crank travel, it will be

apparent that the overlapping of explosions must deliver a very uniform application of power. Eight-cylinder Vee-engines have been devised having the cylinders spaced but 45 degrees apart, but the explosions cannot be timed at equal intervals as when 90 degrees separate the cylinder center lines.

Radial Cylinder Arrangements.—While the fixed cylinder forms of engines, having the cylinders in tandem in the four- and six-cylinder models as shown and also in the eight-cylinder and twelve-cylinder Vee-types have

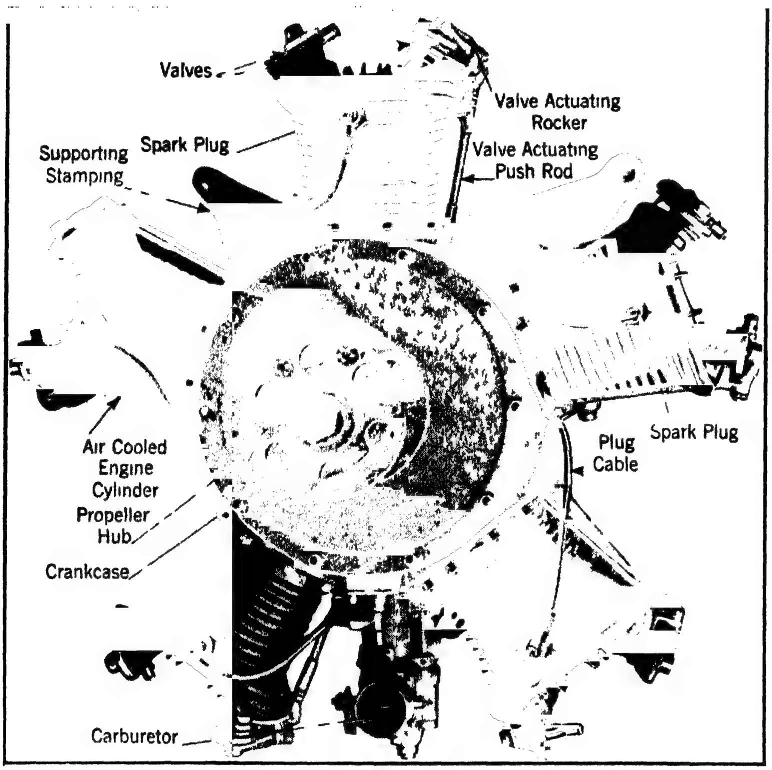


Fig. 78.—The Air-Cat 60 Horsepower Aircraft Engine has Five Radially Disposed Cylinders Cooled by Air.

been generally used and are most in favor at the present time for water-cooled engines, other forms of motors having unconventional cylinder arrangements have been devised, though most of these, other than static radial forms, are practically obsolete. While many methods of decreasing weight and increasing mechanical efficiency of a motor are known to designers, one of the first to be applied to the construction of aeronautical powerplants was an endeavor to group the components, which in themselves were not extremely light, into a form that would be considerably lighter than the conventional design. As an example, we may consider those multiple-cylinder forms in which five cylinders are disposed around a short

crankcase, either radiating from a common center as at Figs. 78 and 79 or of the fan shape shown at Fig. 80. This makes it possible to use a crankcase but slightly larger than that needed for one or two cylinders and it also permits of a corresponding decrease in length of the crankshaft. The weight of the engine is lessened because of the reduction in crankshaft and crankcase weight and the elimination of a number of intermediate bearings and their supporting webs which would be necessary with the usual in-line construction. While there are six power impulses to every two revolutions

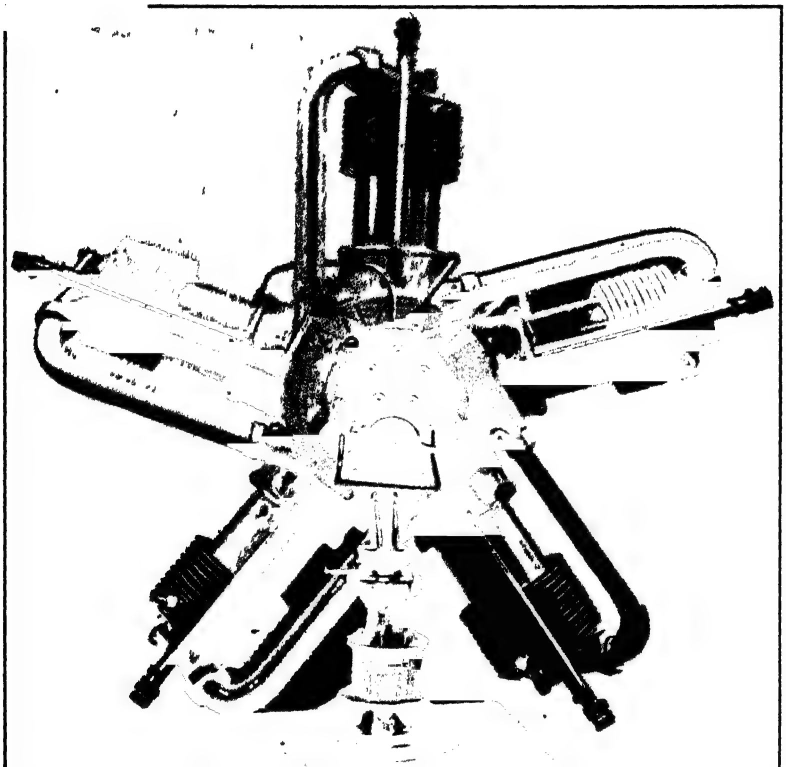


Fig. 79.—Anzani 40-50 Horsepower Five-Cylinder Air-Cooled Engine of Early Design.

of the crankshaft, in the six-cylinder fan engine shown at Fig. 80 they are not evenly spaced as is possible with the more conventional arrangement used by Anzani in which the cylinders are equally spaced and radially placed but staggered so that a two-throw crank is used and the engine is virtually two three-cylinder forms in tandem relation.

In the Anzani form, which is shown at Fig. 79, the crankcase is stationary and a revolving crankshaft is employed as in conventional construction. The cylinders are five in number and the engine develops 40 to 50 hp. with a weight of 72 kilograms or 158.4 lbs. The cylinders were of the usual air-cooled form having cooling flanges only part of the way down the cylinder,

but more modern designs have the circumferential cooling flanges carried down far enough to cool that portion left unfinned in the early design illustrated. By using five cylinders it is possible to have the power impulses come regularly, they coming 144° crankshaft travel apart, the crankshaft making two turns to every five explosions. The balance is good and power

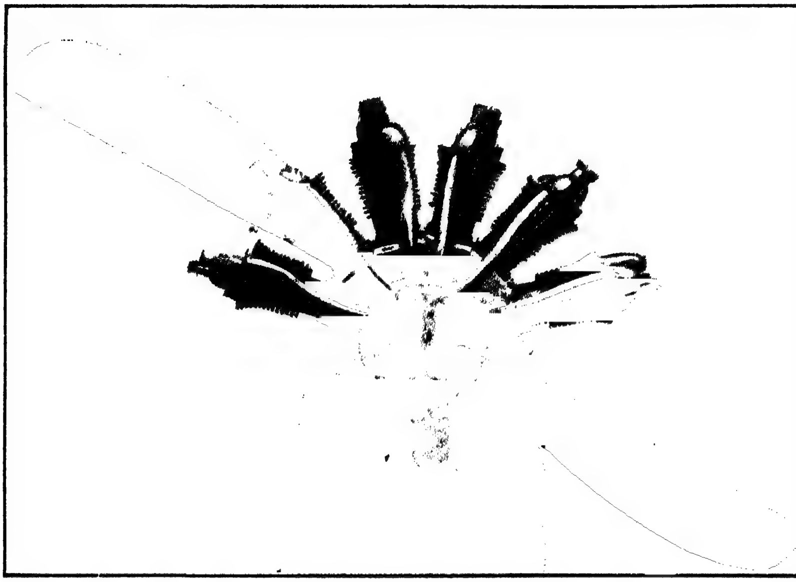


Fig. 80.—Unconventional Early Six-Cylinder Aircraft Motor of Masson Design is Now Obsolete. This Type of Motor was Referred to as the "Fan" Type.

output regular. The valves are placed directly in the cylinder head and are operated by a common push rod. Attention is directed to the novel method of installing the carburetor which supplies the mixture to the engine base from which inlet pipes radiate to the various cylinders. This general type of engine is receiving considerable application to light airplanes in its more modern forms.

In the form shown at Fig. 80 six cylinders were used, all being placed above the crankshaft center line. This engine was also of the air-cooled form and developed 50 hp. with a weight of 105 kilograms, or 231 lbs. The carburetor is connected to a manifold casting attached to the engine base from which the induction pipes radiate to the various cylinders. The propeller design and size relative to the engine is clearly shown in this view. While flights have been made with the Masson engines, this method of construction is not generally followed and has been almost entirely displaced here and abroad by the static radial motors or by the more conventional eight-cylinder Vee-engines. Both of these designs were used over eighteen years ago and would be of insufficient power for the requirements of modern aircraft. They are illustrated so the reader can understand what has been done in this art and to show that the pioneer engine designers realized the advantages of static radial engines at an early date.

Early Rotary Engines.—Rotary engines such as shown at Fig. 81 are generally associated with the idea of light construction and it is rather an interesting point that is often overlooked in connection with the application of this idea to flight motors, that the reason why rotary engines were popularly supposed to be lighter than the others is because they form their own flywheel, yet on airplanes, engines are seldom fitted with a flywheel at all.

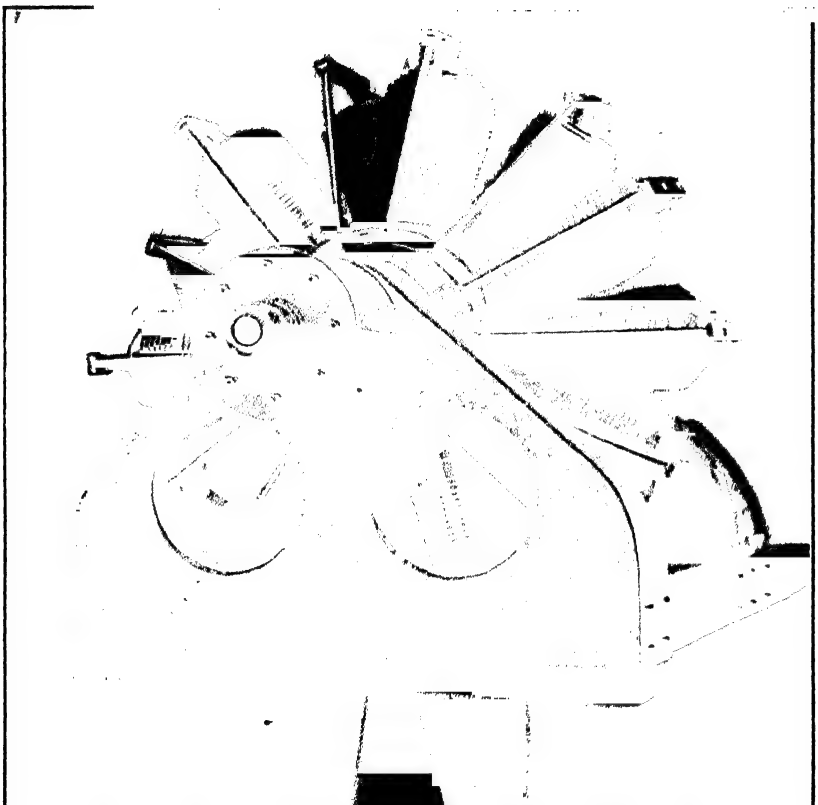


Fig. 81.—The Gnome Fourteen-Cylinder Revolving Cylinder Air-Cooled Motor Was a Good Example of Early Multiple-Cylinder Engine Design.

This type of engine was first introduced in this country about twenty years ago for automobile work and a car, known as the Adams-Farwell was marketed for several years with a rotary air-cooled motor disposed back of the seat, the cylinders turning in a horizontal plane and driving a vertical crankshaft. This American design of Farwell was copied abroad by various designers, the Gnome engine being the best known of the early rotary engines applied to flight service.

As a matter of fact the Gnome engine was not so light because it was a rotary motor, and it was a rotary motor because the design had been adopted as that most conducive to lightness and also most suited to an engine working in this way. The cylinders could be fixed and crankshaft revolve without increasing the weight to any ex-

tent as is done by most engineers today when the static radial is the rule and a rotary cylinder motor the exception. There are two prime factors governing the lightness of an engine, one being the initial design, and the other the quality of the materials employed. The consideration of reducing weight by cutting away metal is a subsidiary method that ought not to play a part in standard practice, however useful it may be in special cases. In the Gnome, Le Rhone and other rotary engines the lightness is entirely due to the initial design and to the materials employed in manufacture. Thus, in the first case, the engine is a radial engine, and has its seven or nine cylinders spaced equally around a crankchamber that is no wider or rather longer than would be required for any one of the cylinders. This shortening of the crankchamber not only effects a considerable saving of weight on its own account, but there is a corresponding saving in the shafts and other members, the dimensions of which are governed by the size of the crankchamber. With regard to materials, nothing but steel was used in the Gnome motor and most of the metal was forged chrome nickel steel, a very expensive construction. The beautifully steady running of the engine was largely due to the fact that there were literally no reciprocating parts in the absolute sense, the apparent reciprocation between the pistons and cylinders being solely a relative reciprocation since both travel in circular paths, that of the pistons, however, being eccentric by one-half of the stroke length to that of the cylinder.

While the Gnome type engine offered numerous advantages, on the other hand the head resistance offered by a motor of this type was considerable; there was a large waste of lubricating oil due to the centrifugal force which tended to throw the oil away from the cylinders and special blends of an unstable organic (castor) oil were required to insure lubrication; the gyroscopic effect of the rotary motor was detrimental to the best working of the airplane, and moreover it required about seven per cent of the total power developed by the motor to drive the revolving cylinders around the shaft.

Of necessity, the compression of this type of motor was rather low, and an additional disadvantage manifests itself in the fact that there was no satisfactory way of muffling the rotary type of motor. The Gnome engine has been widely copied in various European countries, but its design was originated in America, the early Farwell engine being the pioneer form. It has been made in seven- and nine-cylinder types and forms of double these numbers. The engine illustrated at Fig. 81 is a fourteen-cylinder form. The simple or one-bank engines have an odd number of cylinders in order to secure evenly spaced explosions. In the seven-cylinder, the impulses come 102.8° apart. In the nine-cylinder form, the power strokes are spaced 80° apart. The fourteen-cylinder engine is virtually two seven-cylinder types mounted together, the cranks being just the same as in a double-cylinder opposed motor, the explosions coming 51.4° apart; while in the eighteen-cylinder model the power impulses come every 40° cylinder travel. Other rotary motors have been devised, such as the Le Rhone and the Clerget in France and several German copies of these various types. The mechanical features of these motors will be considered later as a matter of general interest though this system of construction is practically ob-

solete at the present time. It showed modern engine designers the way to realize light weight and compactness, however.

Static Radial Engines.—The static radial engine in its various forms, having from five to nine air-cooled cylinders delivering their power on a single crank offer all the advantages of compactness and light weight presented by the fixed crank, revolving cylinder types with the added advantages of not producing high gyroscopic forces in operation nor offering any difficulties as pertain to lubrication or carburetion. They are cheaper to construct as aluminum can be used advantageously in the crankcase

Fig. 82.—Wright "Whirlwind" Engine, an Air-Cooled, Nine-Cylinder Static Radial Type of 200 Horsepower was the Engine Used by Colonel Charles Lindbergh in his Nonstop Flight from New York to Paris, and has a Remarkable Record of Other Accomplishments to its Credit.

whereas it could not be used, owing to the stresses produced, in rotary cylinder engines. The crankcase of the Gnome engine, for example, was made of a forging of chrome nickel steel which had to be machined laboriously from a rough and heavy member to a thin-walled, smooth casing. In static radial engines, aluminum or dural forgings or castings that are approximately of the finished form may be used and much less costly machine work is required. Less material must be removed and it can be removed faster, bringing production costs down.

One of the most popular of the fixed radial air-cooled engines and one that has received a very wide commercial application is shown at Fig. 82, as it looks viewed from the front and in part longitudinal section through

the vertical cylinder at Fig. 83. This Wright "Whirlwind" engine was accorded worldwide recognition because it was used by Colonel Lindbergh in his epochal transatlantic passage from New York to Paris in May, 1927. It is a nine-cylinder type, rated at 220 hp. with an extremely good weight-horsepower ratio. All the important internal parts are shown and the construction can be easily studied. A crankshaft that is no more com-

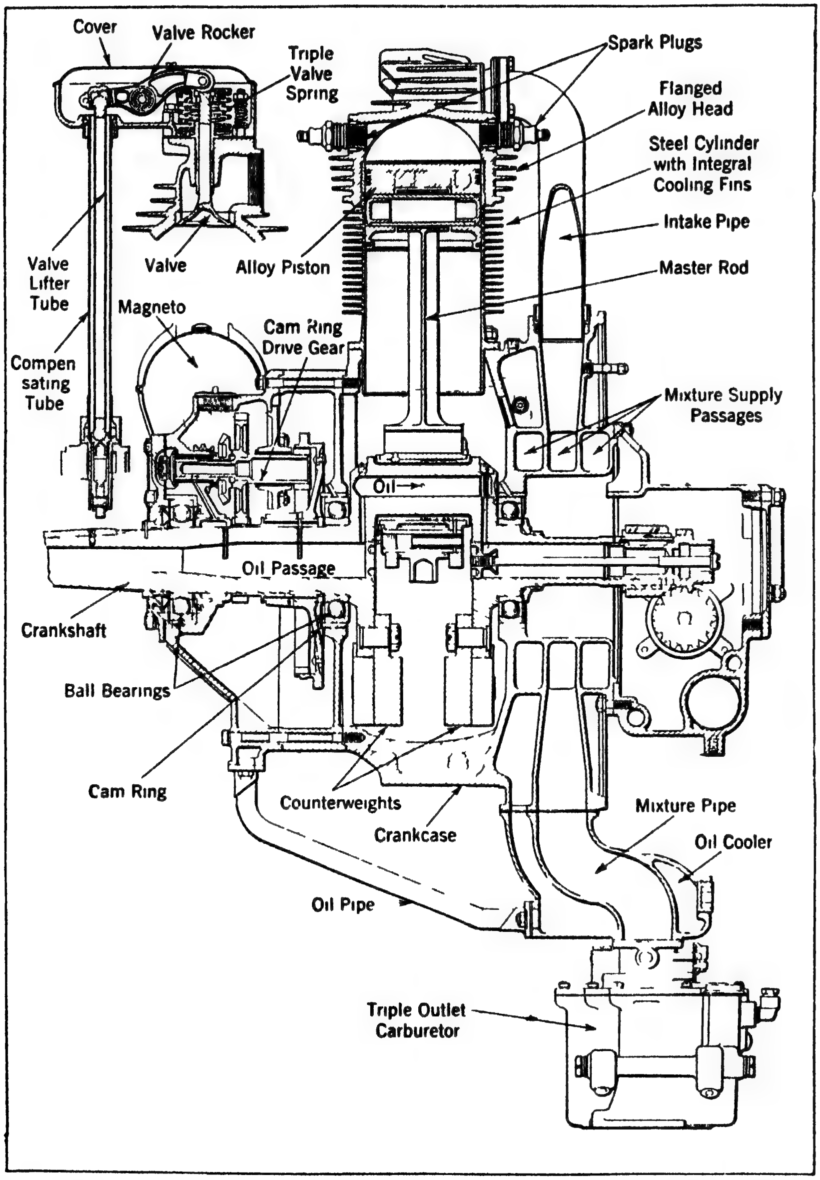


Fig. 83.—Sectional Diagram Showing Internal Construction of Wright "Whirlwind" Motor.

plicated than that required for a single-cylinder engine, as it has but a single crankpin, is used. The master connecting rod big end is provided with eight pins to which eight link rods are hinged. Balance weights are provided to counterbalance the weight of the master connecting rod and insure smooth operation. The cylinder is a steel barrel, provided with cooling fins, threaded and shrunk into a cast aluminum alloy head, also provided with liberal cooling flanges. This is a very common method of construction today and its development is due to the work of Charles Lawrance,

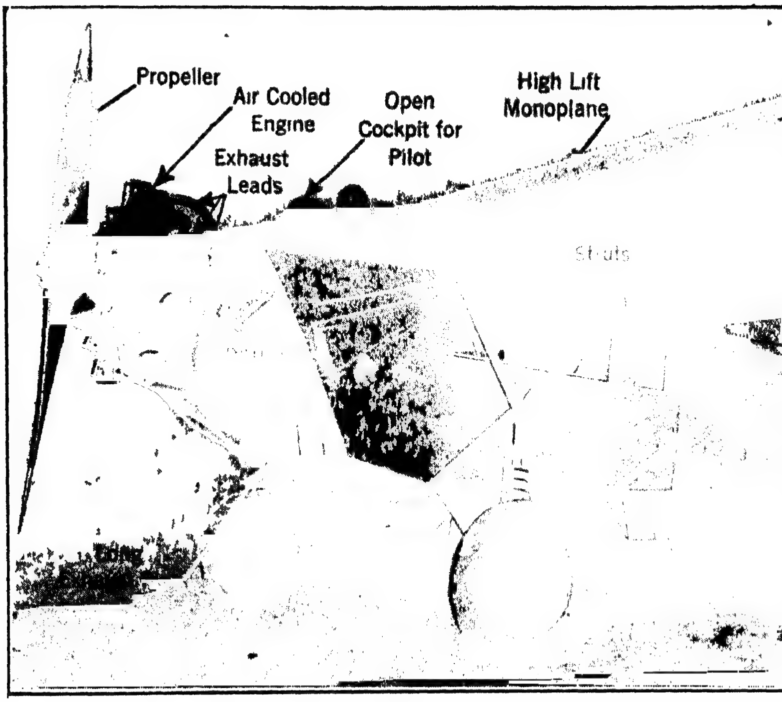


Fig. 84.—The Designer of the Fokker "Universal" Monoplane has Provided an Open Cockpit for the Pilot and a Cabin for the Passengers. Note the Installation of the Wright "Whirlwind" Radial Cylinder Air-Cooled Engine at Nose of Fuselage, and Exhaust Pipe Leading Burnt Gases Out Under the Airplane Cabin.

designer of the engine, in conjunction with army and navy authorities over a period of years. The engine is very compact and can be easily installed at the front of a fuselage, as shown at Fig. 84, or as a wing motor, as at Fig. 85 and as no water-cooling radiators or connections are necessary, the powerplant weight, complete and ready to fly is very considerably less than a water-cooled engine of the same power.

Another very successful static radial engine is shown in part section at Fig. 86. This is the Siemens-Halske, a German design now being manufactured in this country where it is known as the Ryan-Siemens. This is made in five-, seven- and nine-cylinder forms. Attention is directed to the large number of ball bearings used in this engine, which greatly reduce engine friction. The construction and operation of these engines, as well as

other practical types will be considered more in detail in other chapters as the subject is more fully developed.

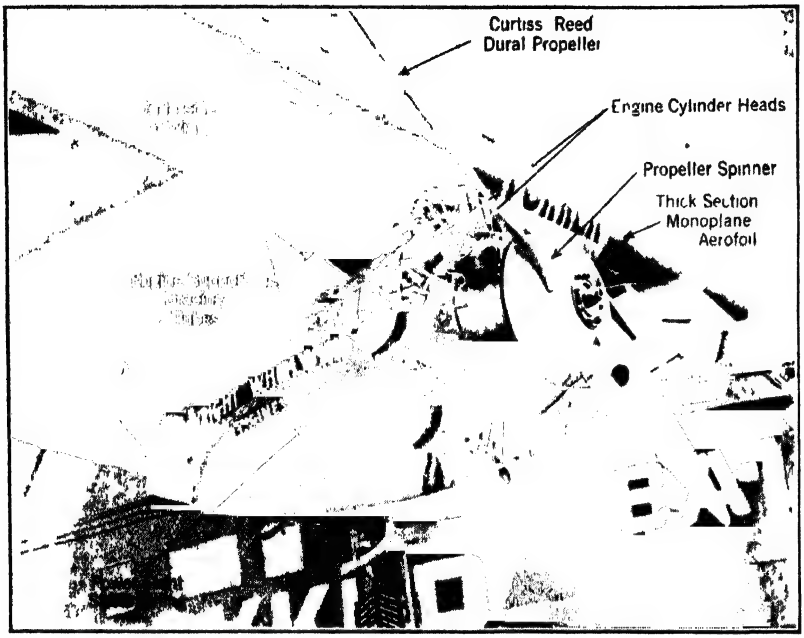


Fig. 85—How the Nine-Cylinder Air-Cooled Wright "Whirlwind" Engine is Installed in a Nacelle for Placing Under the Wing of the Fokker Trimotor Monoplane.

Considerations in Air-Cooled Radial Engine Designs.—The well-known aeronautical engineering authority, Capt. Robert W. A Brewer, considers the problems confronting a designer of radial air-cooled engines in a recent issue of *Aviation* and discusses the matter in his usual interesting and informative manner. Mr. Brewer states that the designer of an air-cooled radial engine for commercial use must decide where his probable market lies. He must study carefully the present production needs and the future trend of development. Together with those aeronautical designers and backers who are open to confer upon the subject a decision will be arrived at as to the general objects such as: Brake-horsepower; speed of revolution; cooling and cowling; weight; number of cylinders; disposition of accessories; lubrication and fuel arrangement; the kind of fuel to be used; mounting, type and arrangement; the selling price of the complete engine; and does this include a starter.

These few main headings provide the designer with much material for a start. He will then probably base his first conclusions upon what can be done for the cost at which the job can be produced. This item, though last on the above list is the overshadowing influence in the problems to be met. This question of costs predominates throughout every consideration, as he must realize that certain standard and desirable forms of accessories must be purchased by the engine maker at "aviation prices." This cuts down considerably the figure of allowable cost of the engine alone. As an

example, a proposition may be submitted concerning a low priced engine design to include many desirable features, but the cost of these accessories would almost amount to the total permissible cost of the job. So, one can readily understand that the minimum of cost is very definitely determined, irrespective of the size of the engine.

We must therefore, choose a size in which the cost of the accessories is not overwhelming in relation to the total. Let us take the brake-horsepower as having a value of x and this is to be developed at N revolutions per minute with a direct drive. The next consideration is the kind of fuel to be used and, for commercial work, we will assume, perhaps rightly so, that commercial gasoline should be the fuel. This will require us to make some sacrifice in the power-for-size output of the engine, but this is warranted in view of the ultimate object. The engine must operate economically with the cheapest fuel that can be bought for the purpose. The fuel question limits the compression ratio desirable, so let this be 4.5 to 1. The details of design will determine the brake mean effective pressure obtained with such proportions, the compression pressure will depend upon

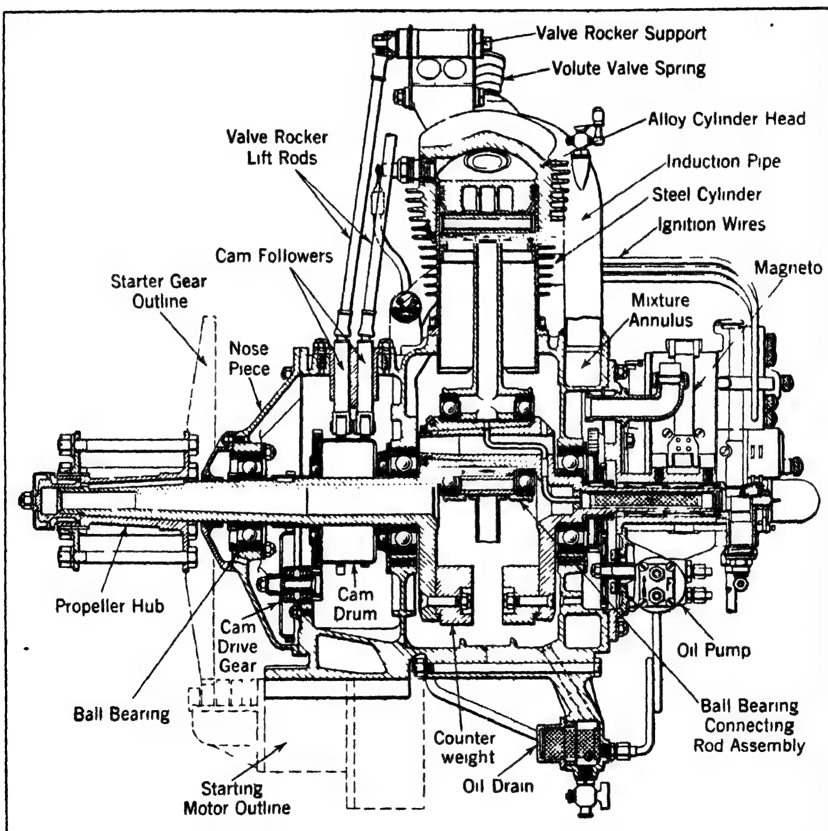


Fig. 86.—Longitudinal Sectional View of the Siemens Static Radial Cylinder Air-Cooled Motor.

many details of the induction system and the valves and may vary between limits as far as fifteen pounds per square inch apart. The brake mean effective pressure will also depend upon the efficacy of the cooling of the cylinder-head, pistons and valves so that for maximum output, all these factors must be carefully worked out.

Distribution of Mixture a Problem.—A problem of major importance in a radial engine is that of the distribution of the explosive mixture to the cylinders, and although this is very vital in all engines, it is particularly difficult in a radial. Any student of radial engines will be acquainted with certain forms of manifolds, duplex, triplex and spiral, which are in use, but the great point is to keep fuel from settling in the lower parts of the manifolds and to deliver it in equal quantity to the upper parts. This can be done satisfactorily by simple means, as has been demonstrated, and the fuel intimately mixed with the air during the process. Economy and absence of fire risk are two benefits derived from such a method, irrespective of the resulting equality of explosion torque.

How, now is the volumetric displacement of the engine to be disposed? First we will make a conservative estimate of its magnitude based upon the performance of similar engines. Divide this up into trial numbers representing possible numbers of cylinders which can be used. In conventional types of air-cooled radials, this number will be odd and limited to five, seven or nine we will neglect three as such a number is not usually satisfactory except for quite small engines. The stroke-bore ratio may have many values and here comes a major problem to solve, what shall it be? Generally speaking, the desirable feature is to provide as much cylinder bore circumferentially as can be crowded round the crankcase. This is limited by piston interference. It will be necessary to make several trial layouts so that this interference can be avoided and at the same time holding the overall diameter of the engine in its complete form within desirable limits. This overall dimension is also governed by the angle of the valves and by the size and type of springs used. The angle of the valves is in turn governed by the diameter required or rather the ratio of the diameter of a valve to that of the cylinder. As usually some compromises have to be made, and although it is better to keep the valve gear truly radial from the mechanical standpoint, this adds to the overall diameter of the complete engine. On the other hand, some change of direction of the motion involves side thrusts on certain parts, either rockers or tappets, which must be suitably designed to cope with the loads imposed upon them.

The ratio of length of connecting rod to crank is fixed within certain limits so that the skeleton scheme is now complete. How shall the connecting rod assembly be made? There are two general classifications in common practice, the divided big end of the master rod with a one-piece crank; the solid big end with a two-piece crank. In the former, a one-piece crankcase can be used, thus avoiding a circumferential joint at the center. In the latter, this is practically a necessity if a bench assembly of the rods is desired. There are adherents of both lines of development and arguments pro and con regarding both. It is possible to make a lighter big end of the single eye type but this is accompanied by the features of the divided crankcase and the divided crank, adding cost to the work. Some divided jobs

suffer from leakage of oil, faulty and non-interchangeable fitting and crank trouble. Mechanically, it is quite feasible to make a single overhung crank of sufficient strength and stiffness and with it to provide a second forward bearing for the propeller support. The rear part of the crank then forms a medium for driving the accessories only. This means three bearings on the crank assembly. For cheapness and in order to obtain a two-bearing crank it would seen inadvisable in Captain Brewer's opinion to use a divided crank as so much depends upon the proper fitting of the sleeve or coupling holding the two parts of the crank together. In the past, failures have resulted here. However, some very successful engines have a divided crank such as the Ryan-Siemans shown at Fig. 86, the Pratt and Whitney "Wasp" and the English Bristol Jupiter, all powerplants of international prominence and excellent reputation.

Supposing a divided master rod is decided upon, a very nice problem arises as to the location of the linkpins. Owing to the peculiar motion of the subsidiary rods, a careful study must be made of the positions of the pistons at various angular positions of the crankpin. The pistons should all come to the proper top center at the proper angular interval passed through by the crankpin. These positions should be laid out and it will be found that the correct locations of the linkpins fall in a group which crowds them together opposite the master rod. Another arrangement is possible where the pins are equally spaced, but this requires departure from standard uniform dimensions of parts. This alternative for a commercial engine is not desirable as one of the main attractions of the radial is the similarity and simplicity of its components.

Piston Design Considerations.—Piston design is governed by the space available and freedom from interference of the piston skirts with one another and with the balance weights. Clearance from certain linkrods must also be watched not only for the pistons but also for the cylinder barrels. These latter generally have to be cut away to permit the linkrods to swing. The clearance in the cylinders opposite the master rod has to be the greatest depth. For the reason that the master rod must be stronger and wider, than the linkrods, its width governs the width of the clearance slots in the cylinder barrels.

The question of balance is one requiring much care. Static balance can be arrived at by the conventional method but this does not relieve the load on the crankpin. The Bristol Company's system, as employed in the Jupiter engine, is very neat. There may be other schemes by which this loading may be relieved. Crankpin loading is one of the limiting factors of the radial engine as regards output for size. Whereas piston design for the "in-line" engine type must provide for piston side thrust, this is of equal magnitude in all cylinders.

Piston Side Thrust Varies.—Radial engines offer the problem of variable thrust due, not only to the different obliquity of the linkrods, but to the interdisposition of thrust in the system of connecting rods. This is brought out very clearly in an analysis of stress, contained in the McCook Field Report 2504 from which is taken the following table:

Piston Side Pressures. Pounds per square inch

Cylinder	During power stroke *maximum*	*mean*
1	119	88
2	82	63
3	89	72
4	106	85
5	131	95
6	154	103
7	159	97
8	147	85
9	118	73
1 on notched side	136	90

No. 1 is the master rod cylinder.

Comparing the magnitude of this thrust with "in-line" engines we find, for the Liberty twelve-cylinder engine, a maximum of 52.5 lb. per sq. in. and a mean of 43.2 lb. per sq. in. during the power stroke. These figures show that the side areas for pistons for radials should be as large as possible to keep the intensity of stress within practical limits. In the particular engine analyzed, the magnitude of the side thrust on the piston carrying the master rod varied from plus 1,520 lb. at 80 deg. crank angle to minus 1,508 lb. at 260 deg. crank angle. This was for a cylinder of 5⅝ in. bore.

Piston Weights Important.—Piston weight must be kept down to limit the inertia forces as these govern the design of the linkrod ends. The problem is to accommodate on the big end of the master rod a group of linkpins and rod ends, which have freedom of movement and yet are spaced as closely as possible to the center of the crankpin. There must be a sufficient factor of safety, proper bearing area and provision for lubrication.

The inertia force to be provided for can be found:

$$F_i = 0.0000284 \times W \times r \times N^2 \times f_a$$

where F_i = the inertia force

W = the weight of the reciprocating part in lbs.

r = the radius of the crank arm in inches.

N = the revolutions per minute

f_a = the acceleration factor which has values as follows for the usual engine proportions:

at top center —1.273	120 deg. after —0.637
20 deg. after —1.149	160 deg. after —0.731
60 deg. after —0.363	180 deg. after —0.727

In designs where equal angular spacing of the linkpins is resorted to, there will result a variation of piston position with equal angular displacement of the crankpin which amounts to as much as twelve degrees. This means that the timing of the ignition and other functions will vary by this amount.

Design of Commercial Cylinders.—Cylinder design and material have been passed over until now, not because of its relative importance but mainly because the question of cost imposes limitations. In such an engine

as we have under analysis, cast iron cylinders have practically no alternative. We must be prepared to carry a little additional weight in order to save a large cost. Cast iron cylinders of a useful commercial size weigh under eighteen pounds each complete with heads, valve pockets, flanges and cooling ribs. Similar cylinders without heads come out at ten pounds, so that by the time aluminum alloy heads are fitted, with valve seats, ports, etc., the saving of weight is very small and the additional cost is high. Practically, therefore, the cast iron cylinder complete must be used, Captain Brewer believes, if the engines are to be sold at lower prices than the expensive dual metal construction makes possible. In designing this part, the foundry and air-cooling problems predominate. Manifolding of the exhaust gases leaves open a field for ingenuity as the cost of pipe bending work is very high. This also applies to the matter of the inlet pipes.

There is a difference of opinion regarding the mounting of the accessories in a purely commercial engine, on one hand there is the desire for magnetos on the nose of the crankcase so as to give certain accessibility. This involves cowling which might otherwise be dispensed with. Others prefer the accessories at the rear of the engine, which introduces some difficulty in drawing the complete assembly through the mounting ring. For commercial work, a starter should be provided. This should be cheap and simple, hand operated for preference. It is possible to combine such a unit with the rear assembly, adding somewhat to the total length of the engine. Regarding magnetos it is just a moot point as to whether two are needed. The magneto is probably as reliable as any other part of the engine and plug trouble might well be met by a single magneto operating two independent sets of plugs. These few problems give some indication as to why the designer has done certain things that may not conform to the views of the critic and why radial engines differ in many aspects of design though approximately the same in cylinder arrangement.

QUESTIONS FOR REVIEW

1. Name five important engine parts and their functions.
2. Why are multi-cylinder engines best for automotive use?
3. Which type of engine, a six or an eight, has the more uniform torque; which is in better balance?
4. Are four-cylinder engines always of the in-line type?
5. Name minimum number of cylinders to secure overlapping impulses.
6. Why is an eight Vee engine better than an in-line type for airplane use?
7. How many common cylinder arrangements are possible for a twelve-cylinder engine?
8. What is the advantage of the W type?
9. What is the main disadvantage to rotary cylinder engines?
10. Why is the static radial engine so popular for airplanes?

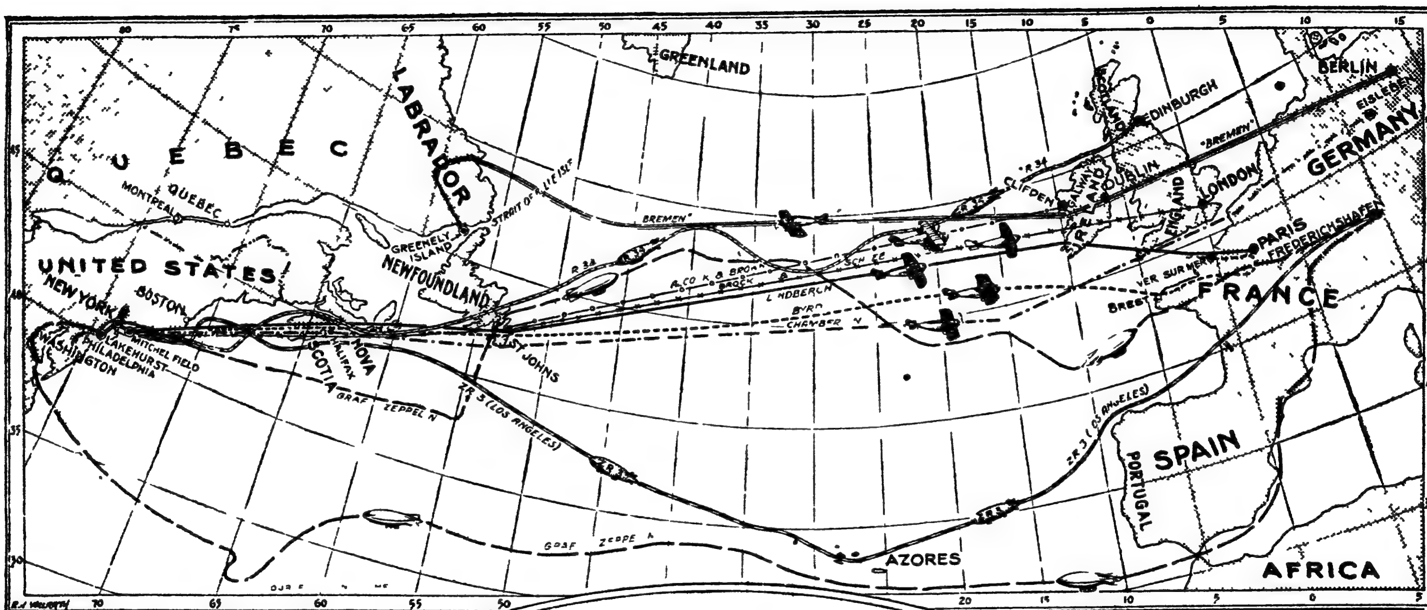


Fig. 87.—Diagram Showing Transoceanic Flights Made by Various Types of Aircraft. All Airplane Flights, Except that of the Bremen, were from West to East. The Dirigibles R-34 and Graf-Zeppelin were the Only Craft to Make Round Trips Up to the End of the Year 1928.

CHAPTER VII

AVIATION ENGINE FUELS—ANTI-KNOCK MIXTURES

Properties of Liquid Fuels—Distillates of Crude Petroleum—Cracked Gasoline—Baumè Gravity—Volatility of Fuel Important—Liquid Fuel Produced from Coal—Alcohol May be Used—Benzol and Similar Fuels—Zeppelin Fuel Gas—Theories of Fuel Knock in Engines—Explanation of Catalytic Action—Theory of Anti-Knock Chemicals—German Anti-Knock Fuel—Peroxides Produce Knocking—Peroxide Formation During Compression—Carbon Formation in' Cylinders—Rate of Carbon Formation.

There is no appliance that has more material value upon the efficiency of the internal-combustion motor than the carburetor or vaporizer which supplies the explosive gas to the cylinders. It is only in recent years that engineers have realized the importance of using carburetors that are efficient and that are so strongly and simply made that there will be little liability of derangement. As the power obtained from the gasoline engine depends upon the combustion of fuel in the cylinders, it is evident that if the gas supplied does not have the proper proportions of elements to insure rapid combustion, the efficiency of the engine will be low. When a gas engine is used as a stationary installation it is possible to use ordinary illuminating or natural gas for fuel, but when this prime mover is applied to automobiles or airplanes it is evident that considerable difficulty would be experienced in carrying enough compressed coal gas to supply the engine for even a very short trip. Fortunately, the development of the internal-combustion motor was not delayed by the lack of suitable fuel.

Properties of Liquid Fuels.—Engineers were familiar with the properties of certain liquids which gave off vapors that could be mixed with air to form an explosive gas which burned very well in the engine cylinders. A very small quantity of such liquids would suffice for a very satisfactory period of operation. The problem to be solved before these liquids could be applied in a practical manner was to evolve suitable apparatus for vaporizing them without waste. Among the liquids that can be combined with air and burned, gasoline is the most volatile and is the fuel utilized by most internal-combustion engines applied to airplanes.

The widely increasing scope of usefulness of the internal-combustion motor has made it imperative that other fuels be applied in some instances because the supply of gasoline may in time become inadequate to supply the demand. In fact, abroad this fuel sells for 50 to 200 per cent more than it does in America because most of the gasoline used must be imported from this country or Russia. Because of this, foreign engineers have experimented widely with other substances, such as alcohol, benzol, and kerosene, but more to determine if they can be used to advantage in motor cars than in airplane engines.

Distillates of Crude Petroleum.—Crude petroleum is found in small quantities in almost all parts of the world, but a large portion of that produced commercially is derived from American wells. The petroleum obtained in this country yields more of the volatile products than those of

foreign production, and for that reason the demand for it is greater. The oil fields of this country are found in Pennsylvania, Indiana, and Ohio, and the crude petroleum is usually in association with natural gas. This mineral oil is an agent from which many compounds and products are derived, and the products will vary from heavy sludges, such as asphalt, to the lighter and more volatile components, some of which will evaporate very easily at ordinary temperatures.

The compounds derived from crude petroleum are composed principally of hydrogen and carbon and are termed "Hydrocarbons." In the crude product one finds many impurities, such as free carbon, sulphur, and various earthy elements. Before the oil can be utilized it must be subjected to a process of purifying which is known as refining, and it is during this process, which is one of destructive distillation, that the various liquids are separated. The oil was formerly broken up into three main groups of products as follows: Highly volatile, naphtha, benzine, gasoline, eight to ten per cent. Light oils, such as kerosene and light lubricating oils seventy to eighty per cent. Heavy oils or residuum five to nine per cent. From the foregoing it will be seen that the available supply of gasoline was largely determined by the demand existing for the light oils forming the greater part of the products derived from crude petroleum. New processes have been recently discovered by which the lighter oils, such as kerosene, are reduced in proportion and that of gasoline increased, though the resulting liquid is neither the high grade, volatile gasoline known in the early days of motoring nor the low grade kerosene or distillate. Special gasoline, known as "aviation gas" is used for most aviation engines.

The low thermal efficiency or high fuel consumption of the prevailing type of automotive engine contributes strongly to the demand for engine fuel, requiring gasoline to be produced in far greater quantities than would be necessary if the fuel were more efficiently utilized. Every gain in thermal efficiency, therefore, means a corresponding increment to the fuel supply. Increased thermal efficiency may also be made to compensate the consumer for such advances in fuel prices as may develop. Thermal efficiency, therefore, has a pivotal bearing upon the quantity and price of engine fuel and though now tending to work to the disadvantage of the automotive industry may be turned to its advantage.

Cracked Gasoline.—As the engine-fuel situation now stands, the most effective means for expanding the supply of gasoline is through rapid development of "cracking" methods of refining whereby gasoline is made from fuel oil and kerosene. But it does not necessarily follow that a continuation of the present degree of dependence upon "cracking" constitutes the most effective means for insuring an ample supply of engine fuel at a price most favorable to the user. When ordinary gasoline is used in a high-compression engine, there is likely to be a "knock" which is undesirable. Cracked gasoline has been found to be less subject to this tendency than is the more highly regarded straight-refinery type. Frequent claims are made that cracked gasoline is superior to the straight-run type when engines are worked under heavy load, and experiments with the phenomena of knocking have tended to substantiate these claims. In general, cracked gasoline is marketed in the form of blends which are not detected by the

average user. It is believed, however, that even if the user were able to distinguish cracked gasoline, it would very likely be on account of its advantageous rather than its disadvantageous behavior.

Baumé Gravity.—Up to the last few years it was customary to rate gasoline on the basis of its Baumé gravity, and even today users are inclined to believe that a high Baumé gravity, which means a low specific gravity, is a desirable property. The specific gravity test, under certain conditions, does give an indication of the distillation range of gasoline, but in general furnishes highly unreliable information in this particular. It

Baume Hydrometer and Specific Gravity Equivalents

For liquids heavier than water (such as electrolytes of storage batteries, electro-plating baths, etc.), at 60° F

$$\text{Specific gravity} = \frac{145}{145 - °\text{Bé}} \text{ and Baumé} = 145 - \frac{145}{\text{Sp. Gr}}$$

Baumé	Specific Gravity	Baumé	Specific Gravity	Baumé	Specific Gravity	Baumé	Specific Gravity	Baumé	Specific Gravity
0	1 000	10	1 074	20	1 160	30	1 260	40	1.381
1	1 006	11	1 082	21	1.169	31	1.271	45	1.450
2	1 014	12	1 090	22	1 178	32	1.283	50	1.526
3	1 021	13	1 098	23	1 188	33	1.294	55	1.611
4	1 028	14	1 106	24	1.198	34	1.306	60	1.705
5	1 035	15	1 115	25	1 208	35	1.318	65	1.812
6	1 043	16	1 124	26	1 218	36	1.330	70	1.933
7	1 050	17	1.132	27	1.228	37	1.342	....	....
8	1 058	18	1.141	28	1.239	38	1.355	....	....
9	1 066	19	1 150	29	1.250	39	1.367	....	....

For liquids lighter than water (gasoline and other engine fuels), the Baumé scale is based on the formula:

$$\text{Specific gravity (at 60°F.)} = \frac{140}{130 + \text{Bé}} \text{ or Baumé} = \frac{140}{\text{Sp. Gr.}} - 130$$

The scale is as follows:

Baumé	Specific Gravity	Lbs. in Gal.	Baumé	Specific Gravity	Lbs. in Gal.	Baumé	Specific Gravity	Lbs. in Gal.	Baumé	Specific Gravity	Lbs. in Gal.
10	1.0000	8.33	31	0.8695	7.25	52	0.7692	6.42	73	0.6896	5.77
11	0.9929	8.27	32	0.8641	7.21	53	0.7650	6.39	74	0.6863	5.74
12	0.9859	8.21	33	0.8588	7.16	54	0.7608	6.36	75	0.6829	5.71
13	0.9790	8.16	34	0.8536	7.12	55	0.7567	6.32	76	0.6796	5.68
14	0.9722	8.10	35	0.8484	7.07	56	0.7526	6.29	77	0.6763	5.65
15	0.9655	8.05	36	0.8433	7.03	57	0.7486	6.26	78	0.6730	5.63
16	0 9589	7.99	37	0.8383	6.99	58	0.7446	6.22	79	0.6698	5.60
17	0.9523	7.94	38	0.8333	6.95	59	0 7407	6.19	80	0.6666	5.57
18	0.9459	7.88	39	0.8284	6.91	60	0.7368	6.16	81	0.6635	5.55
19	0.9395	7.83	40	0.8235	6.87	61	0.7329	6.13	82	0.6604	5.51
20	0.9333	7.78	41	0.8187	6.83	62	0.7290	6.10	83	0.6573	5.48
21	0.9271	7.73	42	0.8139	6.80	63	0.7253	6.07	84	0.6542	5.45
22	0.9210	7.68	43	0.8092	7.76	64	0.7216	6.03	85	0.6511	5.42
23	0.9150	7.63	44	0.8045	6.72	65	0.7179	6.00	86	0.6481	5.40
24	0.9090	7.58	45	0.8000	6.68	66	0.7142	5.97	87	0.6451	5.38
25	0.9032	7.54	46	0.7954	6.64	67	0.7106	5.94	88	0.6422	5.36
26	0.8974	7.49	47	0.7909	6.60	68	0 7070	5.91	89	0.6392	5.33
27	0.8917	7.44	48	0.7865	6.57	69	0.7035	5.88	90	0.6363	5.30
28	0.8860	7.39	49	0.7821	6.53	70	0.7000	5.85	......	....	......
29	0.8805	7.34	50	0.7777	6.49	71	0.6965	5.82	......	....	......
30	0.8750	7.29	51	0.7734	6.46	72	0 6930	5.79	......	....	......

has already been pointed out that a mixture of kerosene and casing-head gasoline might have the same gravity as a straight-run gasoline. In addition it can be stated that the types of crude oil produced in different parts of the country have different physical and chemical properties and that for a given gravity there may be considerable difference in boiling range or vice-versa. Specific gravity cannot always be considered an index of volatility, and it is this factor that is of real importance.

Volatility of Fuel Important.—Volatility is really an important factor and any hydrocarbon fuel must contain a sufficiently large amount of volatile constituents to permit starting when cold. Volatility is different than kindling temperature or igniting point as fuels of the same kindling temperature will differ materially in volatility and in their output of inflammable vapors at normal and even sub-normal temperatures. The difference in volatility between gasoline and kerosene can be shown readily by pouring a little gasoline into one watch-glass and some kerosene into another and trying to light each with a match. Their ignition temperatures are almost identical. It has been determined that kerosene is 20 to 30 deg. lower than gasoline, but this has nothing to do with the fact that kerosene cannot be lighted with a match. There is not enough vapor above the kerosene to ignite; the match flame simply goes out. Of course, the gasoline can be lighted because of the vapor arising from it, which makes it easy to start.

To show the difference in the flames of alcohol and gasoline, we will compare them in the same manner. The alcohol flame is blue. The yellow in the gasoline flame is incandescent carbon. During combustion the oxygen prefers burning the hydrogen to burning the carbon. In burning gasoline in this way the structure is such that the oxygen can readily get at hydrogen enough to satisfy it, and there is not enough oxygen present to burn the carbon completely. In the case of alcohol, an insufficient amount of hydrogen is liberated when the alcohol is broken up to satisfy all the oxygen that can get at it from the air. The result is that the excess oxygen combines with the carbon and the carbon is not left to become incandescent as it is in the case of gasoline.

Such demonstrations are simply to show how these different fuels behave when they are burned in air. They have no particular relationship to the manner in which the same fuel burns when it is first made into a semi-gas and then exploded or ignited from a sparkplug in an engine cylinder while under compression. In this case the results are much different from those when the fuel is spread out and allowed to boil and burn as it chooses. But the tests show the effect of this difference in structure on the behavior of fuels.

Liquid Fuel Produced from Coal.—The possibility of a future shortage of petroleum fuel suitable for automotive engines, however, and of the production of substitutes to avoid such a contingency, is receiving considerable attention in America and Europe. The solution is research in the manufacture of gasoline substitutes from coal, of which enormous quantities remain unmined in this Country. Four known methods of extracting such substitutes from coal are

(1) High-temperature carbonization of coal in by-product coke-ovens or in gas-retorts

(2) Low-temperature carbonization
(3) Hydrogenation of coal
(4) Synthesis of hydrogen and carbon monoxide gases derived from coal, resulting in the production of alcohols

Only the first method is an existing industrial process; the others are in the stages of development. The second and third methods seem to offer important possibilities in the relatively near future, while the synthetic process of producing alcohols from coal-gases is interesting from a theoretical point, as it indicates that hydrocarbons usable in present or slightly modified automotive engines can be produced at moderate cost from inferior coals.

Other products than motor fuels are produced by these processes, such as coke, heavy oils, and gases suitable for illumination and heating. The carbonization methods are dependent economically upon the sale of these in addition to the sale of the gasoline substitutes, but the hydrogenating and synthesizing processes may be self-supporting on the liquid products.

As regards the real possibilities of gasoline substitutes, it now seems probable that oil shale may in the future prove an important source. There are enormous deposits of oil shales in the United States and the extraction of liquid oil from these is undoubtedly feasible, although it has not yet been worked out upon a commercial scale in this country. Oil shale will probably not assume any great importance as a source of engine fuel in the near future, but its potentiality as a resource gives a comfortable sense of assurance that the use of airplanes and other motor vehicles will not have to be discontinued when petroleum resources are exhausted.

Alcohol May Be Used.—Another type of fuel which offers unlimited possibilities for the future and which is already being developed to a certain degree is alcohol. The problems to be solved before this comes into general use are apparently the development of cheaper methods of production and the development of suitable types of engines. Efforts are at present being made to market a fuel containing alcohol and other components, which may be used satisfactorily in present types of internal-combustion engines. A sample of this fuel has been obtained by the Bureau in the retail market and has been subjected to laboratory examination. Its exact chemical composition was not determined, but it was shown to contain both alcohol and benzol as well as a fair percentage of high boiling petroleum naphtha. The sample obtained by the Bureau was being sold at a price somewhat higher than that of engine gasoline. As regards its use, the Bureau has received reports from at least two reputable organizations indicating that it was found at least as satisfactory as ordinary gasoline.

Benzol and Similar Fuels.—The type of gasoline substitute which is of most importance at present is the mixture of hydrocarbons obtained as a by-product in the coking of coal. These so-called coal-tar distillates including benzol, toluol, xylol, etc., are hydrocarbons which are somewhat similar to the hydrocarbons found in petroleum, although of course there are well-recognized physical and chemical differences.

There are several advantages to be gained from the use of benzol, either by itself or mixed with gasoline. These are:

(1) It is a manufactured fuel and can be made anywhere that coking coal is available
(2) It gives slightly more power than gasoline
(3) It gives slightly better mileage per gallon
(4) The running of the engine using it is sweeter
(5) The compression can be raised and the power thereby further increased
(6) There is no "pinking" or detonation
(7) It is cheaper than gasoline in some localities

The opinions of users of this fuel are, however, various, and they even run to opposite extremes. As benzol has no very volatile content, flooding of the carburetor does not help starting from cold, and the only way to get a start on a cold winter day, if heat is not added, is to spin the engine as fast as possible so that the heat of compression vaporizes the benzol. If a mixture of gasoline and benzol is used, there is, of course, a small quantity of the volatile constituent of gasoline which may, in some cases, be sufficient to give the first few explosions. By flooding the carburetor, the quantity of this volatile portion is increased, so that, if a mixture of gasoline and benzol be used, flooding is beneficial. Consequently, in cold weather it is better to use a mixture and to apply heat or to have a small priming tank filled with volatile fuel and used only for starting.

A very necessary warning must be given in regard to this fuel, and that is in connection with its high freezing point. While gasoline may be considered never to freeze, benzol does so at a temeprature above that of ice, 43 deg. Fahr., and the temperature has to be raised very considerably to thaw it again. This is a further argument against using benzol alone in winter, as it may freeze in the pipes or tanks. True, with benzol intended for use as an engine fuel, toluol and other ingredients are supposed to be added by the producers to lower its freezing point, but experience shows that one cannot be sure these additions have been made. The addition of a small quantity of gasoline prevents freezing. Recent tests on aeronautic engines have shown that a mixture of twenty per cent benzol and 80 per cent gasoline is about the best, and is almost as good as straight benzol from the point of view of elimination of pinking. An increase in the amount of benzol beyond this gives inferior results until straight benzol is used. Considering the drawbacks of straight benzol, one comes to the conclusion that a mixture of about the above proportions has no disadvantages if sufficient heat to ensure vaporization can be provided.

Zeppelin Fuel Gas.—Newspaper reports of the transatlantic voyage of the new airship, Count Zeppelin, with Captain Eckener at the helm, stated that the fuel used on the trip to New York, is a mysterious "blue" gas. Experts say that the gas is neither mysterious nor blue. This same gas in a form less pure has been used to light railroad cars in this and other countries for at least a decade, and when used for that purpose has always been referred to as Pintsch gas. It has also been used by farmers and in suburban homes in Europe and America as a fuel for cooking and lighting. Herman Blau of Augsburg, Germany, considered one of the most competent gas engineers of his day, was associated with Julius Pintsch for some time. Pintsch succeeded in manufacturing a hydrocarbon gas which was

so compressible that seventeen volumes of it could be squeezed into one. Pintsch, proud of this achievement, though not as ambitious as Blau, named it for himself. Railroads immediately saw the value of the gas as a fuel for lighting coach interiors, since it could be carried in a relatively small container and was as good, if not better, for lighting purposes than any gas discovered up to that time. Manufacturers of harbor buoys also were quick to seize upon the possibilities and the gas was used extensively in lighting them. Of course, electric lighting is now generally used in railroad coaches.

Blau tried in vain to persuade his friend, Pintsch, to pursue his research, but Pintsch either thought he had reached ultimate success with hydrocarbon gas or was totally indifferent. Blau then made a hydrocarbon gas that, under about 1,800 pounds pressure, with a temperature of minus 50 degrees Fahrenheit, would liquefy, and he gave the product his name. He thought that a much greater quantity could be squeezed into a container than had been possible with the process used by Pintsch. He therefore set out to make some of the lighter hydrocarbons absorb some of the heavier hydrocarbons. He succeeded and produced a gas containing about 1,800 British Thermal Units per cubic foot.

Hydrocarbon is the technical name for the by-products of petroleum such as kerosene and gasoline. After the petroleum is cut for the fourth time, a gas oil is obtained. This Blau used as a base. He used retorts similar in most respects to those employed in the ordinary plant which converts coal into gas; except that they contained iron pipes called vaporizers to keep the oil from coming into contact with the clay retort during the "cracking" process. Much less heat is used under the retorts when hydrocarbons are the base than when coal is. Blau used less in his process than Pintsch did in his—employing oil as a base—because Blau wanted to make a gas that could be liquefied under pressure in a proper temperature. He passed the gas from the retorts through suitable tar extractors, scrubbers, coolers and purifier boxes, and after these processes had a fine quality of oil gas which he passed through a compressor and a cooling device, where it was reduced to a liquid state and put into heavy steel cylinders.

Blau Gas.—Blau gas contains a good many hydrocarbons unsaturated, and because of this fact is a superior fuel for internal-combustion engines. It has a specific gravity of 1.04 to 1.08 and that is one of the main reasons why it appealed to the German Zeppelin Corporation. One of the main difficulties in carrying liquid fuel in a dirigible is that as the tanks are emptied one after the other during the voyage weight must constantly be shifted or water be produced by condensing the exhaust gases of the engines. Blau has recently made claims that he has got his product down to the specific gravity of one, and if this is true the Count Zeppelin should have no difficulty in maintaining her trim and keeping an even keel. The first plant for the manufacture of Blau gas was erected in 1908 in Blau's native city of Augsburg, and others were later built in various European cities. Rights to operate under Blau's patents were obtained by a group of men in this country. Recently a factory has been set up in Friedrichshafen, where the Count Zeppelin was tested. As Blau means "blue" in German,

it was natural to confuse the name of the inventor and "color" of the gas. As the gas is used up, it can be replaced by air which weighs about the same and can be used as a filling for any "ballonets" that may be necessary to keep the lifting gas cells properly distended. Its advantages make it more applicable to lighter-than-air craft than airplanes and the conventional liquid fuels will continue to be used in such machines as do not require gas bag support.

Theories of Fuel Knock in Engines.—The conditions in an airplane engine cylinder are very complex and very different from those found in a laboratory. The only chemical reaction analogous to them is that of flame. Associated with the flame as it passes forward through the cylinder are brilliant light and high pressure. Two waves are produced: a forward or detonation wave and a backward or retonation wave. The detonation wave is the cumulative sequence of increasing temperature, velocity and expansion; the retonation wave, a reaction to the high pressure developed by the detonation wave and sent back in the opposite direction. The three theories of knock are that it is caused (*a*) by the mechanical impact of metallic parts, (*b*) by spontaneous ignition and a simultaneous development of pressure throughout the cylinder and (*c*) by the setting-up of a large detonation wave having large differences of pressure and of wave-front which give rise to vibratory deformations that produce sound.

One theory of the function of knock-preventive material is that it lays down in the cylinder a catalytic agent that will lower the ignition temperatures of other fuels so that they will begin to burn before the flame gets entirely through, and will show the same phenomena throughout. Another theory is that there is introduced in the dope a negative catalyst that slows down the reaction and prevents it from attaining to a detonating velocity. When viewed as a catalyst poison, the function of an anti-knock material is to counteract the catalytic effect of the walls of the cylinder, sparkplug and other substances present within the cylinder, and to enable the reaction to take place at its normal velocity, which is slower than the catalyzed velocity and not sufficiently fast to produce a detonation wave. The fact that certain substances are effective both as catalyst poisons and as knock-preventives and in the same degree indicates that there is a certain parallelism between catalyst poisons and knock-preventives that is worthy of further investigation.

Explanation of Catalytic Action.—Catalysis is defined as the phenomenon that occurs when a substance that apparantly takes no part in a chemical reaction is capable of altering the rate of the reaction; a catalytic agent, as a substance that hastens or alters the velocity of chemical reaction, but after the reaction has been completed is present in its original amount with its original properties. Inasmuch as recent researches on detonation have demonstrated the importance of a careful study of the catalytic action produced in the fuel-mixture by certain compounds, Dr. Schlesinger, Professor of Chemistry at University of Chicago, Chicago, Ill., undertakes to clarify the subject of catalysis in general in an article in the April, 1925 *S. A. E. Journal*, and after showing experimentally various chemical reactions and catalytic effects, discusses from the viewpoint of a scientist the reactions that take place within the cylinder of an automobile,

special reference being made to the detonation and retonation waves that are produced and to knock. Indicating how a study of fundamental facts may lead to hypotheses that may be either verified or disproved, how a speculation may be logically developed into a practical thing, he reasons that a material may be found, which, when deposited on the sides of the cylinder or on the sparkplug, would act as a permanent catalyst, or that an alloy may eventually be discovered from which cylinders may be constructed that will have a continuous catalytic effect on the fuel-mixture.

Two explanations of catalytic effects that have been advanced are (*a*) that the reaction is accelerated by the molecules being absorbed and oriented by the walls of the vessel, and the most active portions of the molecules being arranged in the positions in which they are most easily acted upon; and (*b*) that a catalyst, such as platinum, transforms the molecules from a non-reactive to a reactive condition by absorbing radiations of a particular wave-length and returning to the reaction-mixture those of another wave-length that it is capable of absorbing. Other substances have the property of preventing catalytic action, apparently by poisoning the catalyst, and it is only when these poisons are removed that the reactions become feasible. Among such substances are organic amines, the iodine compounds, the arsenic compounds, the sulphur group, including selenium and tellurium and lead, which is the worst of all. The poisons are supposed to act either by transforming the catalytic agent into a non-reactive compound or by coating it so that the substance to be catalyzed cannot come into contact with it.

Theory of Anti-Knock Fuels.—A new theory of the action of anti-detonating preparations is advanced in a report issued by the American Chemical Society concerning researches covering a large number of chemical compounds. These researches which were carried on in the chemical laboratory of Ohio State University by William Hale Church, Edward Mack, Jr., and Cecil E. Boord, showed that compounds of lead are the best antidotes against knocking in automobile engines. The new theory offered by the investigators attempts to explain the way in which tetraethyl lead prevents knocks: In the explosion which takes place in the engine, tetraethyl lead is decomposed suddenly into infinitesimal particles of metallic lead which act as centers for partial burning. These little particles themselves burn as the flame front approaches them and thus they make the flame travel faster than if they were not present. This condition is described as somewhat like millions of unimaginably small sparkplugs that ignite the gas just ahead of the flame front.

"Thus by virtue of the multiple centers of high temperature created by the burning of these little particles of lead," says the report, "there is initiated evenly ahead of the main flame front a partial oxidation or an auxiliary burning tending to maintain combustion in a region of fuel which otherwise would be subject to detonation.

"The decomposition temperature of anti-knock materials, taken in conjunction with the temperature of the cylinder gases, thus determines at what stage in the cycle they shall begin to function. If the decomposition temperature is low, partial oxidation will begin earlier in the cycle and

extend throughout a larger volume of unburned fuel than if it is high. In the extreme of the latter case, it would cause a lowering in the efficiency of the compound, while in the former it might cause slight pre-ignition.

"The ideal anti-knock compound should possess a decomposition temperature which will cause it to begin to function just with or just after ignition of the charge by the sparkplug."

Five properties, according to the researchers, are essential to this anti-knock material. The first is volatility, with the boiling point under 400 deg. C. The second is that the amount of free metal liberated when the compound is heated in the air should be complete. The decomposition temperature should be between 200 and 300 deg. C. Temperatures developed by oxidation of the metal should be high compared to ignition temperature of the fuel. The particles should be of colloidal size to favor rapid oxidation. The knock in an engine, it is explained, is supposed to be due to the fact that burning of a part of the gas mixture so compresses the unburned portion that this unburned portion becomes hot enough to ignite spontaneously. The little particles of lead distributed throughout the gas mixtures vastly increase the swiftness of the flame travel and make it possible for the flame to reach every part of the gas before it has had a chance to ignite spontaneously.

Value of Anti-Knock Chemicals.—Important to chemical science, it was said, was the finding by the investigators of a method of comparing one anti-knock with another. On the basis of this disclosure a table of values was constructed with tetraethyl lead as the basic compound. The researchers determined and classified all anti-knock compounds as well as compounds without effect in attacking knock in automobile engines. Tetraethyl lead was found to be the most formidable of the anti-knock compounds, and all comparison was based upon a value of 100 attached to this compound as "the anti-knock coefficient."

COMPARATIVE VALUES OF ANTI-KNOCK COMPOUNDS
(Based upon lead tetraethyl as 100)

Lead tetraethyl	100	Bismuth triphenyl	18.2
Lead diphenyl dimethyl	97	Stannic iodide	12.8
Lead diphenyl diethyl	93.5	Tin diethyl diiodide	12.3
Lead diphenyl diiodide	80	Lead thioacetate	8.4
Lead diphenyl dichloride	72	Lead ethyl xanthogenate	7.1
Lead diethyl dichloride	67	Antimony triphenyl diiodide	4
Lead tri-p-xylyl	64.7	Stannic chloride	3.5
Lead diphenyl dibromide	60	Titanium tetrachloride	2.7
Lead tetraphenyl	59	Titanium tetraiodide	2.7
Bismuth trimethyl	20.2	Triphenyl arsine	1.4
Bismuth triethyl	20.2		

German Anti-Knock Fuel.—A new anti-knock fuel marketed under the trade name of Motalin and depending upon iron penta-carbonyl for its anti-detonating qualities has made its appearance in Germany. The fuel as it goes into the tank usually consists of about one part iron penta-carbonyl, Fe $(Co)_5$, to from 400 to 500 parts ordinary gasoline. As this is the first time iron penta-carbonyl has been used commercially as an anti-knock compound considerable interest attaches to a description of the results which have been obtained with it, as given in a recent issue of

Auto-Tecknik from facts supplied by the manufacturers. Iron penta-carbonyl is a yellowish-red fluid of a specific gravity of 1.45 which boils at 217 deg. F. and solidifies at four degrees F. The vapor pressure at 68 deg. F. is about 30 mm. of mercury column, and it therefore evaporates uniformly with the gasoline. It is miscible with most of the organic solvents —with gasoline in all proportions. If protected against light it is a very stable material; when exposed to sunlight it is decomposed gradually—more slowly when exposed to diffused light—iron nona-carbonyl being deposited in gold-colored crystals. When kept in the dark it can be stored indefinitely.

This iron penta-carbonyl shows the surprising property of strongly influencing combustion. Figs. 88 A and B show the effects of a quite small addition of iron carbonyl to an air-hydrogen mixture. The gaseous mixture was enclosed in a glass tube four feet long and was ignited by an electric spark. The flame spreads symetrically upwardly and downwardly. The photographic records were obtained by means of a movable plate upon which equal-spaced time marks were made simultaneously with the combustion by means of a helium tube.

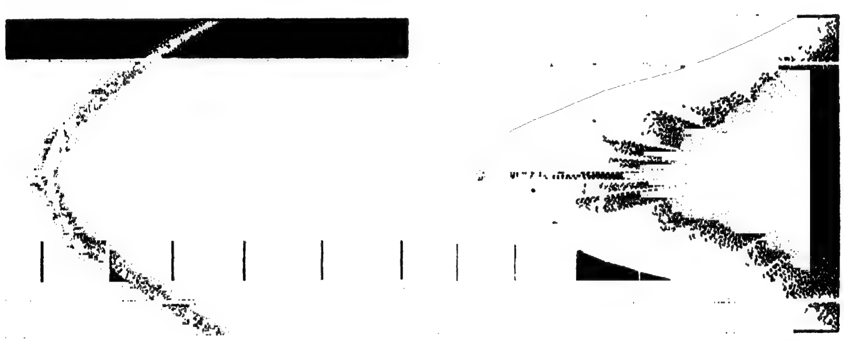


Fig. 88.—Diagram Showing Flame Propagation. A—Record of Flame Using Untreated Fuel. B—Nature of Combustion with Motalin.

Fig. 88 A shows the flame of a gasoline mixture without iron carbonyl addition; Fig. 88 B shows it with this addition. The influence on the flame propagation is clearly shown. Whereas in A the flame-propagation proceeds in a shot-like manner and is completed after 2½ time periods, with the treated fuel the combustion is slowed up to such an extent that the same distance has been covered only in 5¼ time periods. The narrow strip of light in A also indicates a very rapid combustion, while in B the slower combustion produces a wider luminous band. This experiment, therefore, shows the influence of iron carbonyl quite clearly. Knocking of the engine is due to the fact that the normal combustion changes suddenly into detonation; the gaseous waves which now impinge upon the cylinder walls with enormous speed produce a metallic sound, and impermissably high pressures are produced. Thus the elimination of knocking, considered superficially, is the same phenomenon as the slowing up of the combustion in the experiment described.

Peroxides Produce Knocking.—Another theory of detonation and of the action of dopes or anti-detonating compounds has been formulated as the

result of extensive research work carried out in the Air Ministry Laboratory at the Imperial College of Science, London, under the direction of R. O. King, by E. W. J. Mardles, W. J. Stern and N. R. Fowler. The experimental work is described in detail and commented on in a series of articles on "Dopes and Detonation," by H. L. Callender, published in recent issues of *Engineering*. From the results of the experimental work the conclusion is drawn that, with paraffin fuels and ether, detonation is due to the accumulation of peroxides in the nuclear drops during rapid compression. While the peroxides are not formed in quantity sufficient to produce by themselves the detonation observed, they act as a primer, causing simultaneous ignition of the drops. The metallic dopes act by reducing the peroxides as fast as they are formed and preventing their accumulation, thus delaying ignition of the drops.

The following items of evidence are put forward to support this theory:

1. It is shown that at temperatures such as exist in engines at the end of the compression stroke slow combustion occurs (chemical changes take place) in mixtures of detonating fuels with air.
2. It is pointed out that peroxides can be formed by the direct combination of fuel molecules with oxygen molecules, thus obviating the need for the preliminary breaking up of molecules into atoms, which necessitates quite high temperatures.
3. Many references are cited to the effect that organic peroxides have autoxidising and highly detonating characteristics.
4. It is shown that metallic dopes raise the temperatures at which slow combustion begins in fuel mixtures, hence they decrease the activating temperature, that is, the difference between the temperature existing in the combustion-chamber at the end of the compression stroke and that at which slow combustion begins.
5. Metallic dopes decrease the formation of aldehydes in paraffin-air mixtures in slow combustion experiments and promote their formation in alcohol-air mixtures; hence the previous theory that the aldehydes observed among the products of slow combustion of paraffin-air mixtures are oxidation products of alcohols is untenable.
6. Peroxides were detected among the products of slow combustion of detonating fuels, but none was detected when nondetonating fuels were subjected to the same temperatures.
7. Adding organic peroxides to the fuel used in a variable compression test engine lowered the maximum compression ratio which could be usefully employed.
8. When a paraffin (undecane) air mixture was passed through a high compression engine without being ignited, peroxides and aldehydes were found in them, the proportions of both increasing as the compression ratio was increased from six to eight.
9. Tests in a slow combustion apparatus with fuel mixtures of the same kind and proportion showed that more peroxide was formed when the fuel passed through in the finely divided liquid form than if it had previously been vaporized.

The organic peroxides are known to detonate with great violence and

to induce autoxidation, that is spontaneous ignition at atmospheric temperatures. Tests with the apparatus already described showed that when different fuel mixtures were heated up to 300 deg. C., peroxides occurred in the case of detonating fuels, such as paraffins and ethers, whereas none or only a trace could be detected in the case of nondetonating fuels, such as alcohols and aromatics. The addition of metallic dopes to the mixtures of detonating fuels inhibited the formation of peroxides. However, detonation takes place only some time after ignition, when the unburnt mixture has been compressed to a much higher temperature. It is, therefore, concluded that at the time detonation starts the activating temperature excess would be of the order of 300 deg. C. "This suggests that the hypothesis of the formation of some active ingredients during compression is worthy of serious consideration in spite of the shortness of the time available."

Some test results are quoted in *Automotive Industries* showing that in the same engine with the same fuel and the same temperature conditions the highest useful compression ratio is greater the higher the engine speed. This ratio for the engine on which the tests were made rose from 4.45 at 1,200 r.p.m. to 5 at 1,600 r.p.m. The explanation offered is that as the speed is increased a shorter time is available for the formation of these assumed compounds, and the less therefore their effect. If detonation is caused by intermediate products of combustion produced by the activating excess temperature during the latter part of the compression stroke, the effect of dopes can be explained if they can be shown to increase the temperature of initial combustion, thus reducing or eliminating the activating temperature excess. It was shown by experiment that an addition of one per cent of iron carbonyl raises the temperature of initial combustion of undecane 150 deg. C., and smaller additions substantially proportionate amounts.

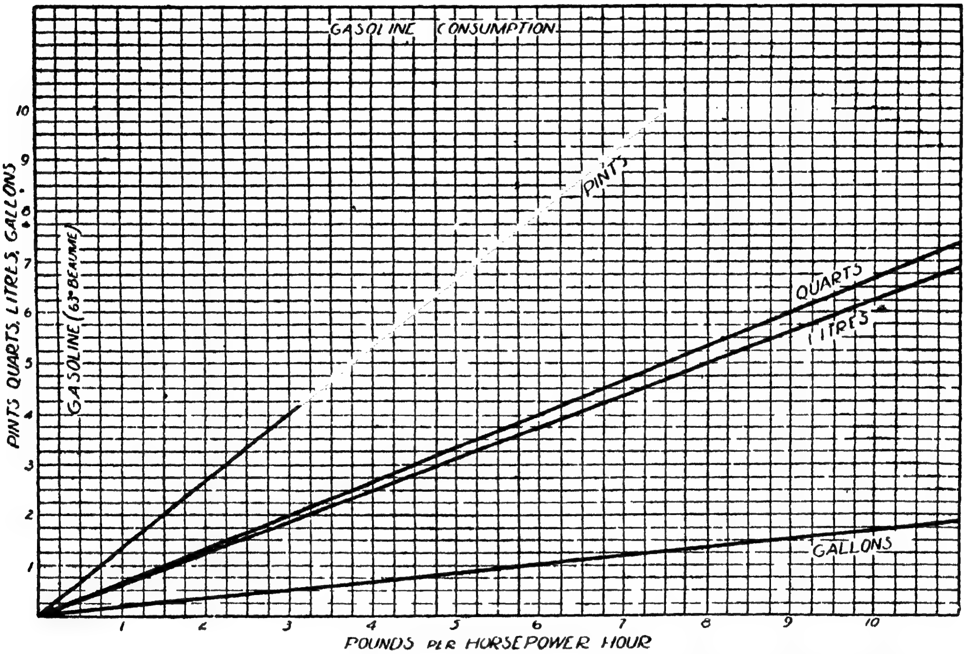
Peroxide Formation During Compression.—Since the formation of peroxides in fuel-air mixtures during the compression period depends chiefly on temperature, the temperatures occurring in engines during the compression stroke are of importance. These are somewhat difficult to determine, but experiments with optical indicators on a single-cylinder test engine at the Air Ministry Laboratory led to the figures given in the following table:

Compression Ratio	Temp. at Beginning of Compression	Temp. at End of Compression
3:1	169	377
4:1	133	384
5:1	112	407
6:1	102	429
7:1	95	452
8:1	90	479

The rather surprising drop in temperature, with increase in compression, at the beginning of the compression stroke is explained by the reduction in the quantity of hot gases remaining in the combustion-chamber and by the reduction in the quantity of waste heat due to the higher thermal efficiency, with increased compression. It is concluded from these figures that in a high compression engine rather more of the fuel is in the form of liquid drops at the beginning of the compression stroke. With the higher compressions the fuel remains more in the form of liquid drops,

which promotes the formation of peroxides. Tests made on an engine running on a gasoline-benzol mixture and then suddenly switched over to undecane while the ignition was shut off, showed slight peroxide and aldehyde contents in the mixture passed through the engine with a compression ratio of six to one, marked contents with a ratio of seven to one and still more marked contents with auto-ignition and detonation with a ratio of eight to one.

Carbon Formation in Cylinders.—In the early days of motoring, before the perfected carburetion and lubrication systems of the present day were evolved, engine operation was largely dependent upon deposits of carbon, which formed rapidly due to imperfect burning of the cylinder contents, oils with a low flash and fire point and, strange as it may seem, to the



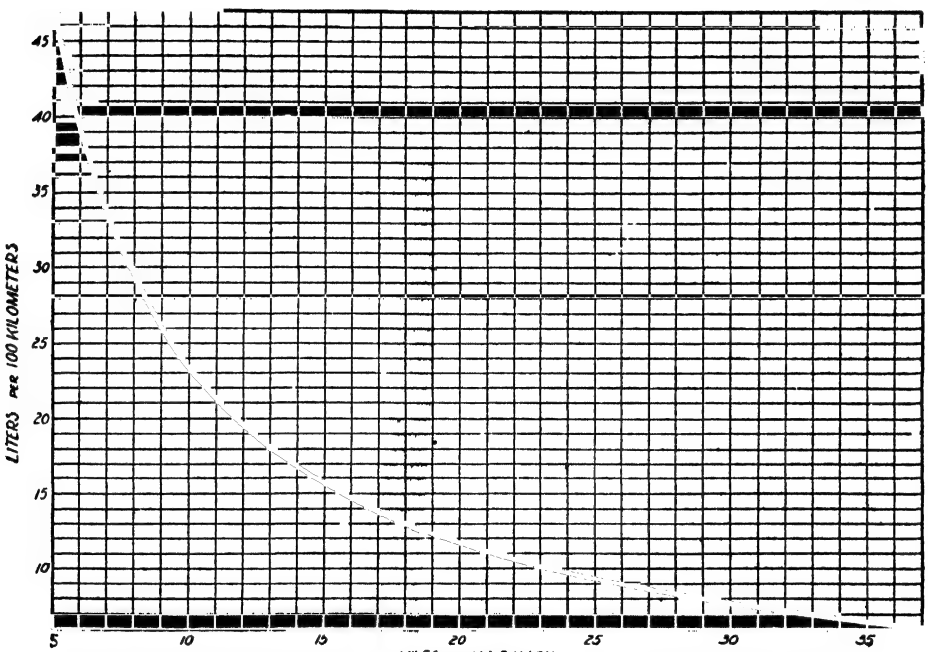
(Society of Automotive Engineers)

CONVERSION OF LB. PER H. P. H. OF GASOLINE INTO PINTS, QUARTS, GALLONS AND LITRES

entrance of quantities of road dust. Analysis of carbon deposits from auto engines showed that over 50% of the formation could be traced to material sucked in through the carburetor and this foreign substance was earthy material such as road dust. Of course, the use of air cleaners has materially changed this. Carbon is formed in airplane engines, also, but this is not of exactly the same nature as that formed in automobile engines because there is considerably less earthy matter in it as there is very little dust in the upper air.

Carbon formation in automotive engines has become a subject of increasing importance in recent years because of the progressively lowered quality of commercial gasoline but this is not a serious problem in airplanes using aviation gas. The fuel of today has a much greater tendency to

detonate than that used in former years. Moreover, it dilutes the crankcase oil, thus indirectly increasing the deposition of carbon. This carbon, acting probably as a heat insulator, results in conditions that favor detonation. Thus, in two ways the lowered quality of present-day gasoline has made carbon deposition a factor of economic importance in the operation of automotive engines. Two classes of compounds for combating this lowered quality of fuel have been marketed. One class is asserted to suppress the detonation, the other class to prevent or remove carbon deposits. While some of those in the first class are of real value, tests show a large number to be worthless. Others, although of real value in suppressing detonation, are too costly to be of economic importance. Those in the second class, with very few exceptions, have been found to be worthless. Some have no



(Society of Automotive Engineers)

CONVERSION OF MILES PER GAL. TO LITRES PER 100 KILOMETERS

effect on carbon deposition. Some even increase it. The regrettable conclusion from testing a number of such compounds is that they were produced and marketed without having been subjected to even a good road-test, much less to a conclusive laboratory-test.

Donald R. Brooks, in detailing some results of tests made with various types of engines to determine the rate of carbon deposition and its influence on engine operation in the *S. A. E. Journal* states that up to a certain point, carbon deposits are not detrimental to engine operation.

The general effects of excessive carbon in the engine are too well known to need detailed discussion. By inducing detonation, it results in over-

heating, loss of efficiency and loss of power when power is most needed, as, in an automobile, when climbing a long hill. To avoid this detonation, *compression-ratios must be kept low, resulting* in continuous lowered efficiency. In short, carbon has been painted as black as it looks.

Four factors are believed to control the formation of carbon in an engine, namely, (*a*) quantity of oil that reaches the combustion-chamber, (*b*) quality of the oil, (*c*) rate of break-down of the oil in the chamber, and (*d*) time. Secondary factors are important only insofar as they influence these primary factors.

Certain substances are shown to accelerate or retard the rate of formation of carbon. Carbon is shown to increase materially the indicated thermal efficiency of an engine operating under conditions such that no detonation or pre-ignition occurs. The increase of efficiency is found to be proportional to the weight of the carbon deposit.

Tests were run with a mixture 30 per cent richer than that giving maximum power, so it is evident that the carbon which may be attributed to the fuel itself is certainly less than ten per cent of the normal total quantity of carbon. Indeed, certain tests that are not here presented indicate that the carbon which may be attributed to the fuel is less than two per cent of the total. Hence it may be stated as proved, that engine factors influence the formation of carbon only insofar as they influence either the character and quantity of oil supply or the temperature in the region of decomposition of the oil. Rich mixtures result in the formation of much more deposit than do lean mixtures, because the unvaporized fuel that is present in the rich mixture dilutes the cylinder-wall oil-film and results in increasing the quantity of oil that reaches the combustion-chamber. Certain substances, when introduced into the combustion-chamber either in solution in the fuel or in the lubricating oil, have the property of changing the quantity of the deposit and, in a majority of cases, its composition as well.

Rate of Carbon Formation.—Examination of the cylinder-head showed that there was an approximate time at which "flaking," or mechanical loss of the deposit by cracking-off, begins. As this flaking becomes more pronounced, deposition and flaking probably tend toward equilibrium, that is, after a long period of engine operation the weight of the deposit practically ceases to increase. Although the data at hand are too meager to allow of mathematical proof of this and it cannot be demonstrated that the carbon weight would approach a definite value, yet to all practical purposes it is evident that an equilibrium weight is reached which is dependent upon the conditions of operation. Thus, in one case, with a lean mixture and a light load, 607 hr. of operation produced 7.79 grams of carbon, whereas, under heavier load and with a richer mixture, 150 hr. of operation gave 22.35 grams of carbon. While in each case the deposit of carbon was still increasing slowly, it seems evident that the light-load operation would never produce as much carbon as the heavier load had produced in 150 hr. This is an important consideration in connection with aviation engines which operate under heavy load conditions practically all the time a plane is in flight. It was noted, also, that whenever the cylinder-heads, upon being removed, showed oil above the pistons, the deposit of carbon in the cylinders was unusually large in quantity. Oil-scraping rings were then

fitted carefully, in the belief that excessive oiling caused the irregular deposition. Tests made with these new rings showed a much smaller deposit, so small in fact that it was evident that the errors introduced in scraping would be an undoubtedly large percentage of the total error.

The following is believed to be indicated by the tests as fact:

(1) Carbon formed in internal-combustion engines arises from thermal decomposition and oxidation of the lubricating oil

(2) Factors that influence the rate of deposition of carbon are those which affect the quantity and character of the lubricating oil that reaches the combustion-chamber and its rate of break-down in the combustion-chamber

(3) The total quantity of carbon that would be formed in unlimited time is dependent upon the conditions of engine operation

(4) An increase in carbon deposit increases the indicated thermal efficiency of an engine and the gain in efficiency is proportional to the increase in the carbon deposit

(5) An increase in absolute humidity in the air-fuel mixture appears to increase the indicated thermal efficiency of an engine.

QUESTIONS FOR REVIEW

1. Why are liquid fuels ideal for airplane engines?
2. What is the most common fuel and how is it obtained?
3. Can fuels be used obtained from other than mineral sources?
4. What is "cracked" gasoline?
5. What are the important characteristics of a good fuel for airplane engines?
6. When is gaseous fuel practical for aircraft?
7. What is "blue" gas?
8. Outline various theories of the causes of fuel knock in engines.
9. What is the best anti-knock chemical?
10. What causes carbon formation in engine combustion-chambers?

CHAPTER VIII

FUEL SUPPLY SYSTEM—PRINCIPLES OF CARBURETION

Liquid Fuel Storage and Supply—Wasp Fuel System—Fuel System for Liberty Engine—Fuel Systems for Long Flights—Vacuum Fuel Feed—Vacuum Boosters—Electrical Fuel Pumps—Barlow Fuel Pump—Air Service Typical Fuel Feed—Principles of Carburetion Outlined—Air Needed to Burn Gasoline—What a Carburetor Should Do.

The problem of gasoline storage and method of supplying the carburetor is one that is determined solely by design of the airplane. While the object of designers should be to supply the fuel to the carburetor by as simple means as possible the fuel supply system of some airplanes is quite complex. The first point to consider is the location of the gasoline tank. This depends upon the amount of fuel needed and the space available in the fuselage or in the airplane wings.

Liquid Fuel Storage and Supply.—A very simple and compact fuel supply system is shown at Fig. 89 A. In this instance the fuel container is placed immediately back of the engine. The carburetor, which is carried by

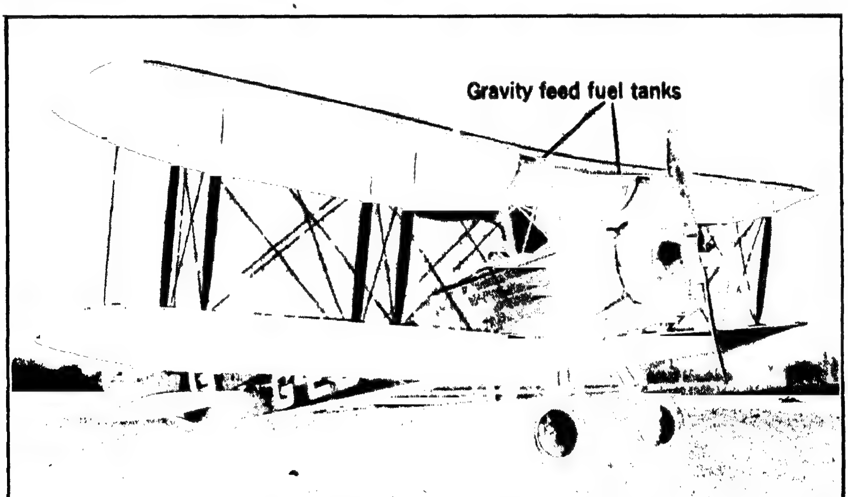


Fig. 89B.—Avro Biplane with Fuel Tanks Located Under the Top Wing at Each Side of the Center Section.

a manifold as indicated, is joined to the tank by a short piece of copper or flexible rubber tubing. This is the simplest possible form of fuel supply system and one used on a number of excellent airplanes. Another system is to have a tank in the center section of a biplane upper wing or high wing monoplane. In the Avro training biplane shown at Fig. 89 B the fuel tanks are carried each side of the center section, and are attached to the under surface of the top wing. The method of carrying the fuel tank in the Pitcairn airplane shown at Fig. 90, back of a fireproof bulkhead and high

enough to permit gravity flow is typical. The engine installation with its carburetor placed below the crankcase and well below the bottom of the tank is shown at Fig. 91.

As the sizes of engines increase and the powerplant fuel consumption augments, it is necessary to use more fuel, and to obtain a satisfactory flying

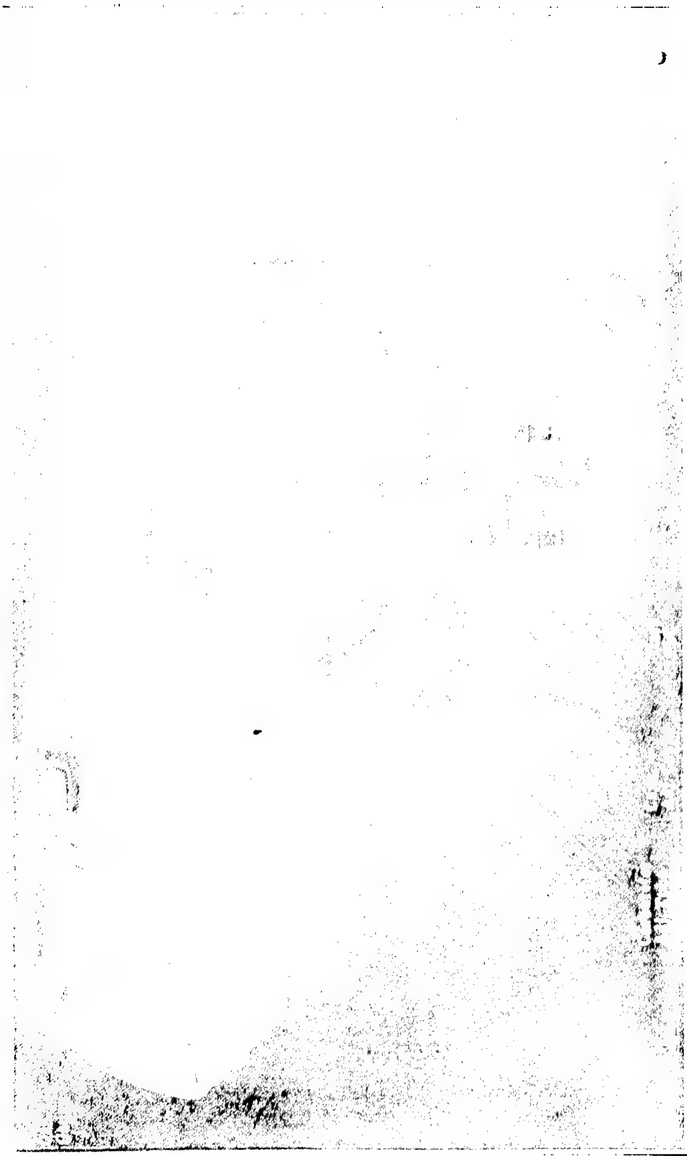


Fig. 89A.—How Gravity Feed Fuel Tank May be Mounted Back of an Automobile or Airplane Engine to Secure a Simple and Short Fuel Supply Line.

radius without frequent landings for filling the fuel tank it is necessary to supply large containers. The way this is done in the Vought Corsair Navy type plane is shown at Fig. 92. The fuel tanks are placed on the sides of the fuselage and form a continuation of the streamlining.

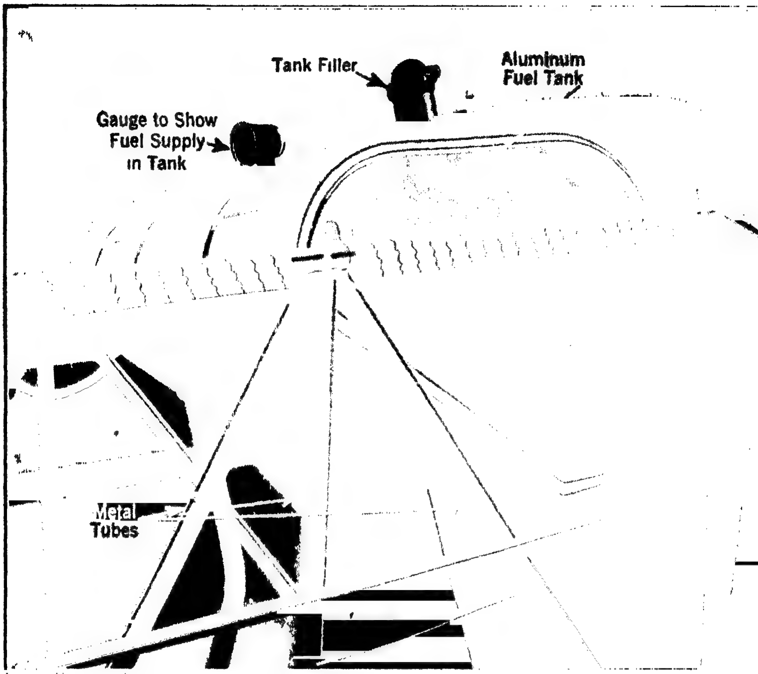


Fig. 90—How Fuel Tank is Mounted in Pitcairn Airplane to Secure Gravity Feed to a Carburetor Placed Below the Engine and Mounted in Front of the Tank.

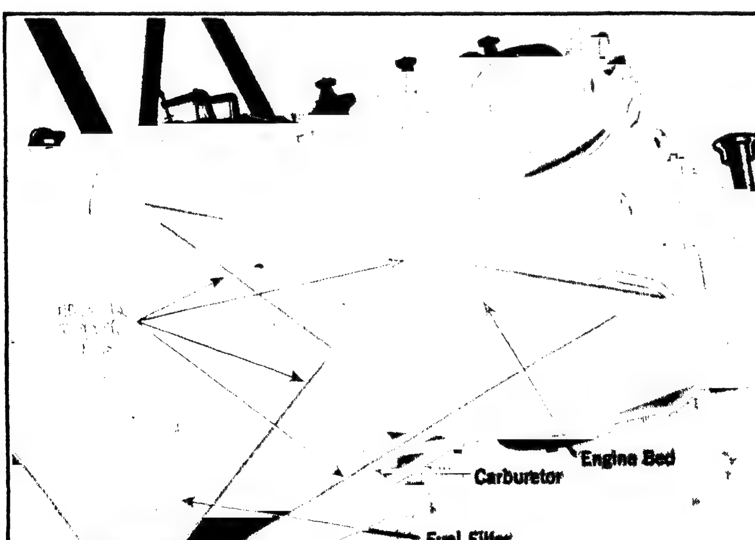


Fig. 91.—Installation of Curtiss OX5 Motor in Pitcairn Airplane Showing Alloy Steel Support Tubes for Laminated Wood Engine Bed Pieces. Note Carburetor Placing at Rear of Engine and Just Forward of Fuel Tank in the Fuselage.

When a very powerful powerplant is fitted, as on battle planes of high capacity, it is necessary to carry large quantities of gasoline. In order to use a tank of sufficiently large capacity it may be necessary to carry it lower than the carburetor. When installed in this manner it is necessary to force fuel out of the tank by air pressure or to pump it to a gravity feed tank because the gasoline tank is lower than the carburetor it supplies and the gasoline cannot flow by gravity from the main tank as in the simpler systems. While the pressure and gravity feed systems are generally used in airplanes, it may be well to describe the vacuum lift system which has been

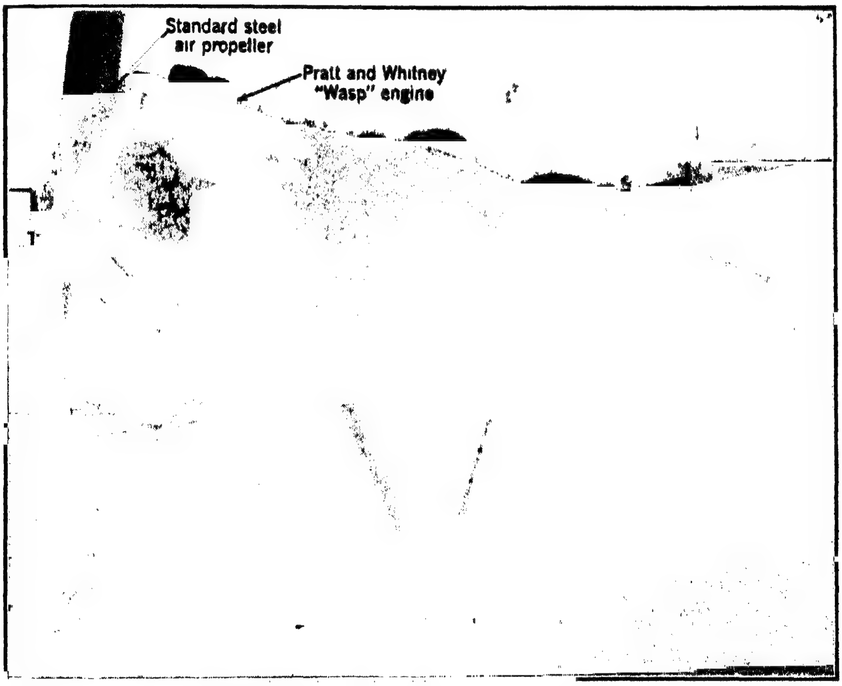


Fig. 92.—Three-Quarter Side View of the Fuselage of Chance Vought Corsair Airplane Used by the U. S. Navy, Showing Installation of Main Fuel Tanks at Side of Fuselage, Where they are Easily Accessible for Filling or Inspection.

widely applied to motor cars and which may have some use in connection with airplanes as certain types of these machines are developed that will be used for purely commercial flying.

Wasp Fuel System.—The Pratt and Whitney Aircraft Corporation recommend the fuel system shown at Fig. 93 for use in connection with their "Wasp" engine. The carburetor is a Stromberg Type NAY-7A. The use of a Pratt and Whitney fuel pump is recommended, whether or not the fuel tank is above the level of the carburetor. One-half inch diameter tubing is recommended for all fuel connections, and in no case should smaller than ⅜ in. diameter tubing be used except for the safety drain from the fuel pump gland. The latter is ¼ in. and must be carried down and out of the cowling without sharp bends and traps and be cut off square. The piping

required includes a supply pipe from the main tank to the fuel pump and a return pipe from the fuel pump back to the tank as shown in piping diagram. A pipe from the fuel pump to the carburetor is supplied. There is a 1/8 in. pipe tap in the carburetor which can be used for connecting to the fuel pressure gauge. A six-pound gauge should be used. A strainer is incorporated in the carburetor, and all fuel should also be strained when filling the tank. In case it is desired to dispense with the use of a fuel pump, the bottom of the fuel tank should be at least four feet above the carburetor with the airplane tail down, which calls for a center section mounting of the fuel tank or a wing mounting if the engine is installed in a biplane.

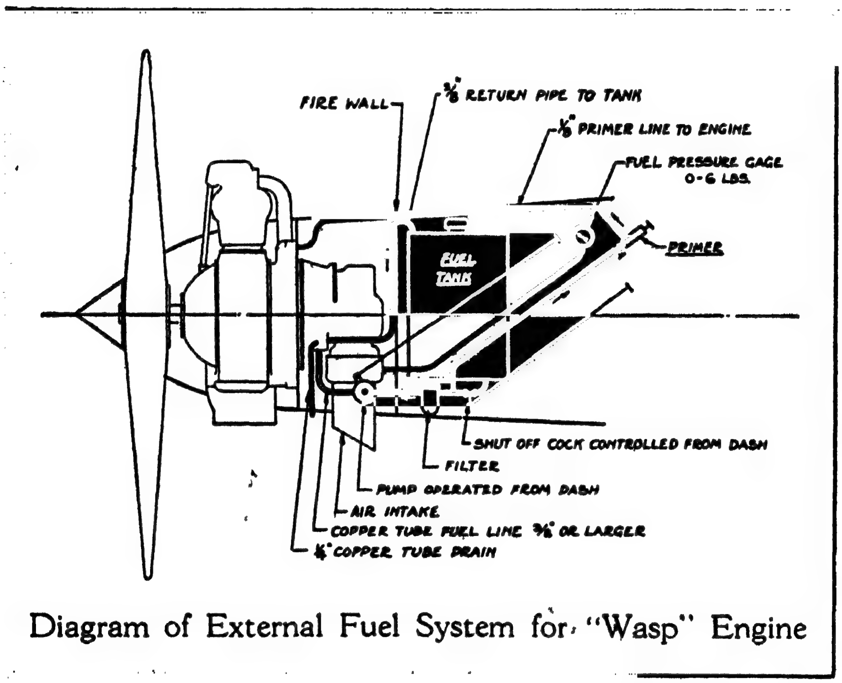


Fig. 93.—Diagram of External Fuel System for Pratt & Whitney "Wasp" Engine.

The primer furnished with each engine can be mounted on the instrument board or near the starting crank, and connected to the primer piping which is already on the engine. The fuel supply for the primer can be taken from any convenient point, preferably at the lowest place in the line, but certainly where fuel is always available. A shut-off cock must be included in the primer supply line at the primer pump, to prevent fuel from entering the engine through the primer system except when priming.

Fuel System for Liberty Engine.—The fuel system utilized in early D H 4 airplanes using the Liberty engine for power is shown at Fig. 94. This was an air pressure system using an auxiliary tank in the center section. The fuel could be drawn from either the main tank, where it was displaced by air pressure produced by an engine driven air pump, or from the auxiliary tank by gravity, depending on the position of the cocks in

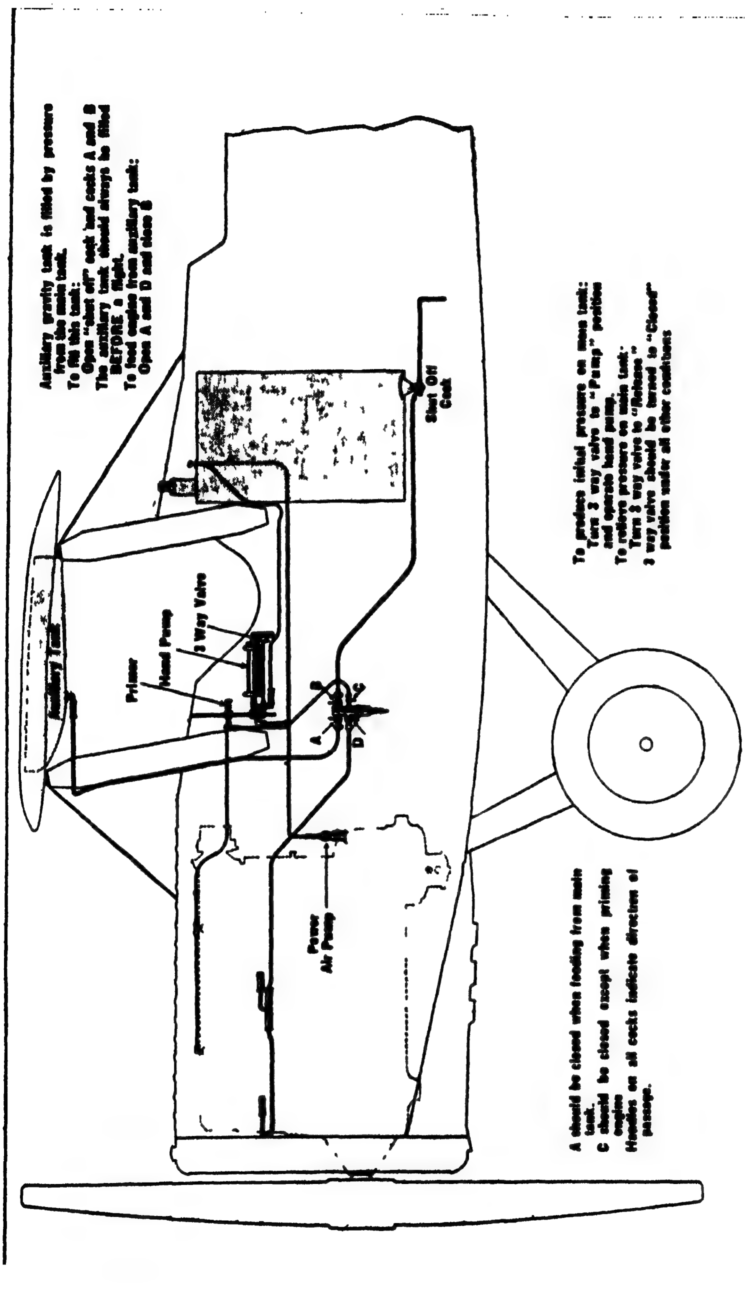


Fig. 94.—Fuel Supply System of Early Model DH4 Airplanes Using Liberty Engines, in which Gasoline was Displaced by Air Pressure in Main Tank.

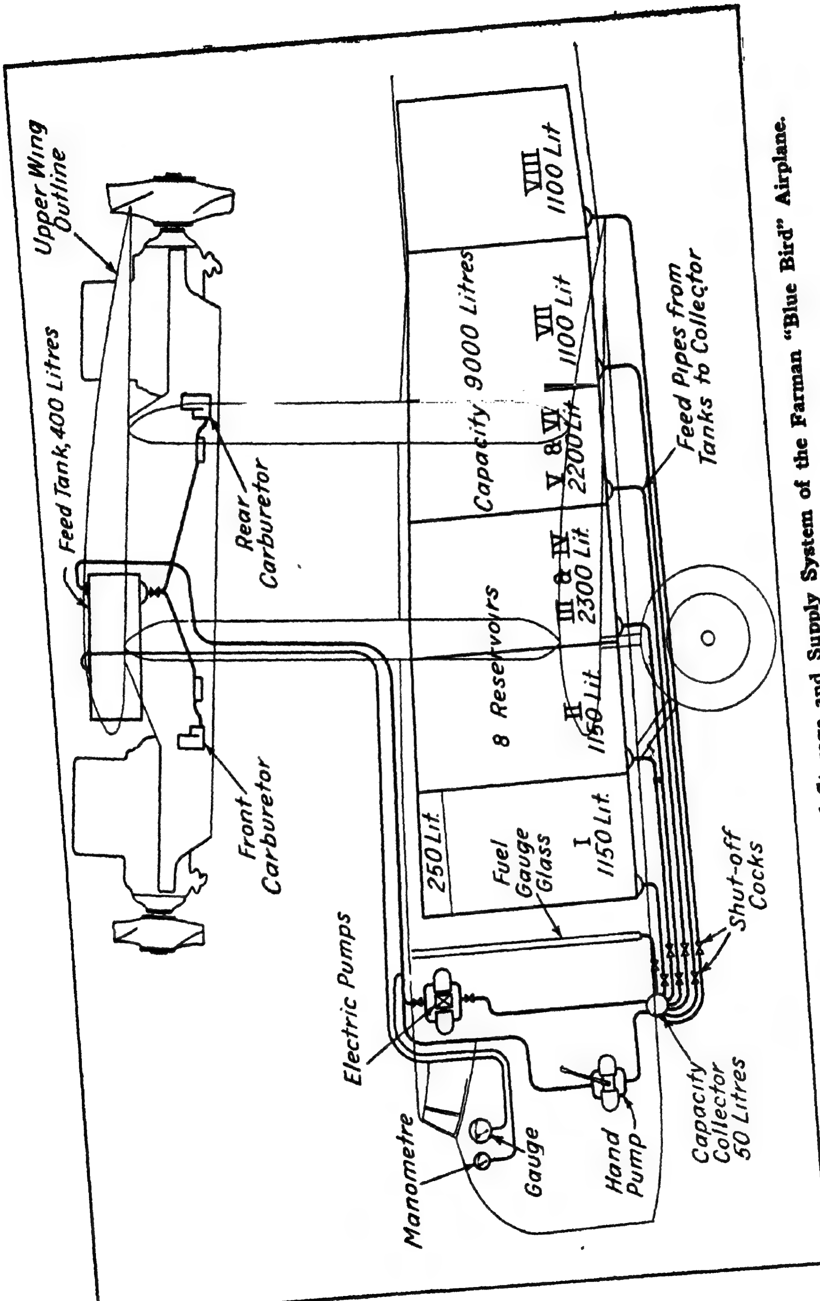


Fig. 95.—Diagram Outlining Fuel Storage and Supply System of the Farman "Blue Bird" Airplane.

the fuel lines. The arrangement can be easily understood by study of the diagrams.

Fuel System for Long Flights.—When airplanes are prepared for long flights, the problem of storing the large amount of fuel required is one that calls for considerable study. The arrangement of main tanks in the Farman Bimotor "Blue Bird" and their relation to the feed tank carried in the motor nacelle depending from the top wing is shown at Fig. 95. The amount of fuel carried was 9,000 liters (2,250 gals.) in eight main tanks and 400 liters (approx. 100 gals.) in the feed tank. Feed pipes lead from each main tank to a collector tank, each feed pipe being controlled by a shut-off cock so one or all can be joined to the collector tank which has a

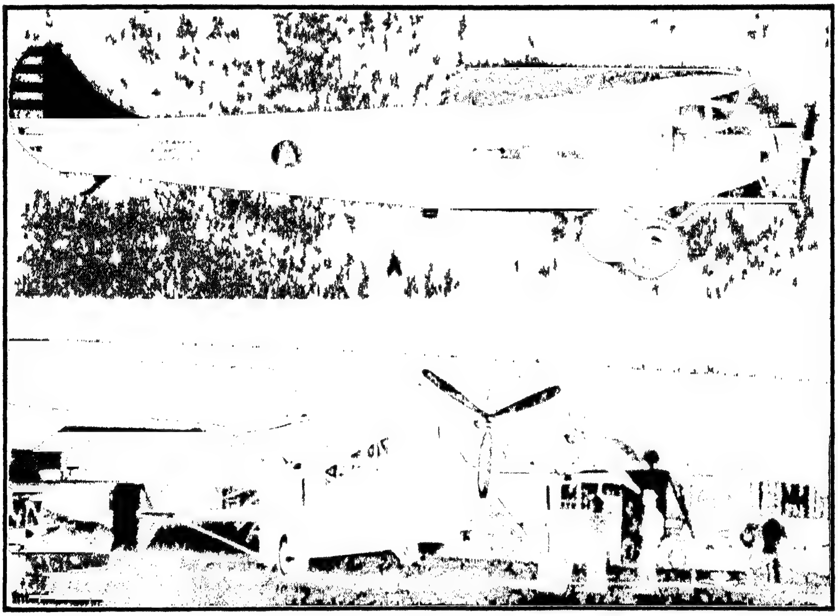


Fig. 95A.—Showing the Record Breaking Army "Question Mark" Trimotor Monoplane. B is a Front View of the Fokker Showing the Three Wright Whirlwind Motors Used for Power.

capacity of slightly more than twelve gallons Two pumps are connected in multiple between the collector tank and the pipe feeding the upper supply tank. One of these is an electric pump, the other is a hand pump. As two engines are used, one driving a tractor screw, the other a pusher, two feed pipes branch off of the feed tank outlet, one going to the front engine carburetor, the other feeding that of the rear engine. The electric pump is depended on to keep the supply tank filled, a gauge indicating the amount available in the fuel supply tank at all times. The hand pump is used for starting and for emergency service. This is usually of the type known as a "wobble" pump.

Refueling in Flight Proven Practical.—Although it has broken every record for sustaining human beings above the surface of the earth, either in balloons, dirigibles, or heavier-than-air craft, the Army's Fokker Tri-

motor Atlantic C2A monoplane "Question Mark" is in no sense of the word a "stunt" plane. It is a true flying laboratory; planned to test the durability of "Whirlwind" engines in continued operation while in the air, to test the plane, its equipment, and its crew for the effect of long continued flight; but most particularly planned to test the practicability of regular refueling of planes while in full flight and with present equipment. Following the record attempt, Major Carl Spatz, commanding officer of the endurance flight, stated that the chief thing they had accomplished was to refuel so many times and under so many varying conditions that no one could doubt the complete success of refueling methods which have been developed. The excerps which follow were taken from a complete descriptive article that appeared in the January 19th, 1929, issue of *Aviation.*

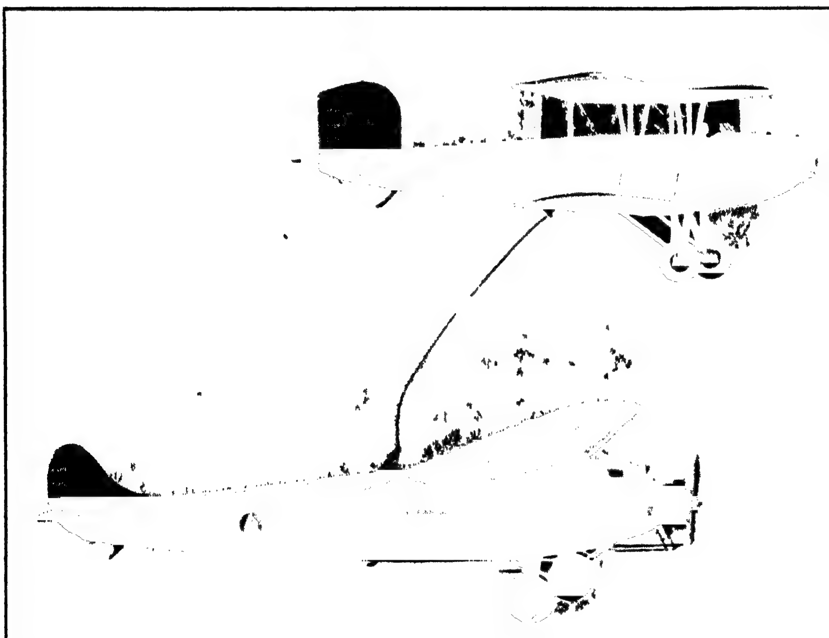


Fig. 95C.—Showing Hose Connection Between Refueling Plane and "Question Mark" When Transferring Fuel.

It is interesting to know that 37 contacts were made during the 150-hour flight between the "Question Mark" and her nurse planes. The 37 contacts entailed a total of approximately four hours of contact flying. During this four hours of contact more than 5,000 gallons of gasoline, 250 gallons of oil, and approximately 2,000 pounds of food and supplies were transferred to the "Question Mark." This totals almost 42,000 pounds of weight handled or almost 21 tons of material that was placed on board the tri-engined Fokker while in flight. Numerous night contacts were made, some of them despite low visibility and bad air currents; and on one occasion a seven minute contact was made when all lights on the "Question Mark" were out due to low batteries, 180 gallons of gasoline being placed aboard. Nineteen full meals were transferred to the crew, while still warm, and

more than two dozen quarts of ice cream were placed on board while still cold. Telegrams, letters, a collapsible bath tub, a supply of bath towels, woolen underwear, a rubber suit for Major Spatz, a window for the cabin, to replace one that had blown away, and many other items of miscellaneous nature were delivered to the crew. It is quite apparent that refuel and supply methods are now feasible enough to keep a plane up indefinitely if engines could be kept turning over.

The crew on the "Question Mark" consisted of Major Carl Spatz, in command; Captain Ira C. Eaker, second in command and chief pilot; First Lieut. Harry A. Halverson, pilot; Second Lieut. Elwood R. Quesada, pilot; and Staff Sergt. Roy W. Hooe, mechanic.

The only official records to be credited to the plane are two new refueling records, one of the new American record for having surpassed the 37 hours, 15 minutes and 40 seconds flight of Lieuts. Lowell Smith and Paul Richter over San Diego, Calif., in an Army De Haviland during 1923, and the other a new world's record for endurance by means of refueling because of excelling the 60 hours, 7 minutes flight of Adjutant Louis Crooy and Sergeant Victor Groenen, made in Belgium during 1928. The new American and world refueling endurance record now stands at 150 hours, 40 minutes 14 seconds, the start having been made at 7:26:46 A.M., January 1, 1929, and the landing at 2:07:01 P.M., January 7, 1929. Both start and finish were on the runways of the Los Angeles Metropolitan Airport, Van Nuys, Calif., under the supervision of N. A. A. officials.

In addition to breaking all distance records the "Question Mark" surpassed the following records for endurance, but since they are in a different class she cannot claim any of them. American Endurance without refueling, Brock and Schlee, 59 hours, 1928; World's heavier-than-air, Johann Risticz and Wilhelm Zimmerman, 65 hours, 25 minutes; Germany, Spherical balloon endurance, Kaulen of Germany, 87 hours; Graf Zeppelin, 111½ hours on 1928 flight to America; French dirigible "Dixmude," 118 hours, 41 minutes.

Some Advantages of Aerial Refueling.—A very practical point is that by refueling often, a transport plane may carry less fuel and more pay-load. By eliminating intermediate landing the wear and tear on the plane is greatly lessened and the element of danger which always enters into a landing is reduced in proportion as the intermediate stops are eliminated. That aerial refueling will greatly speed passenger transport was demonstrated by two refuelings of the "Question Mark" during her flight to the west coast. Over Dallas, Texas, an aerial refueling was accomplished without delaying the plane more than seven minutes, which were used in circling the field and contacting the nurse plane. Continuing on to Tucson, Arizona, the "Question Mark" landed for refueling by a good ground crew which was prepared to refuel quickly, yet the operation consumed 40 minutes, counting the time lost in landing and taking-off. This is a clear gain of more than half an hour, in a fair test, on just one operation.

Having so definitely proved the advantages of aerial refueling, even with present crude equipment, Major Spatz visions the time in the near future when designers will build large transport planes especially for refueling and repair of engines in mid-air. These planes would probably be of 200 or 300 feet wing spread, it is thought, with engines entirely housed

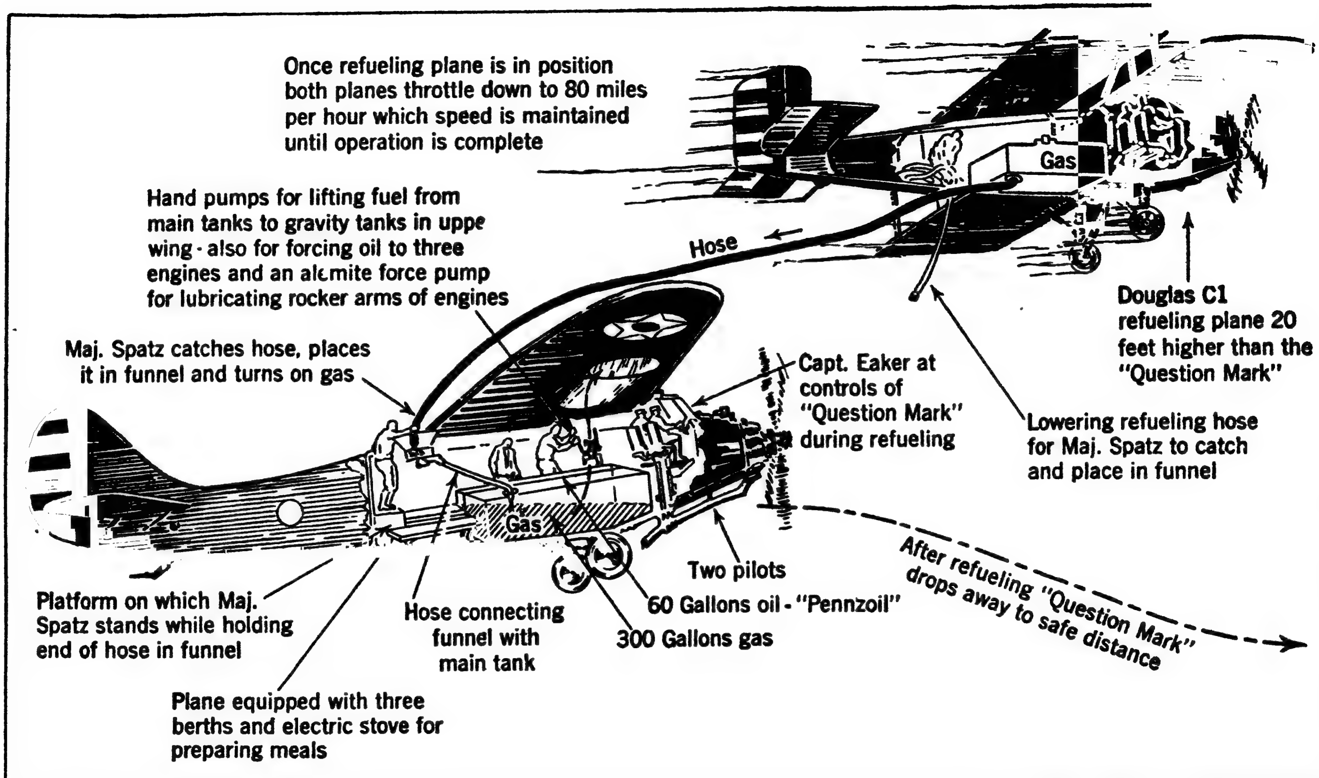


Fig. 95D.—Diagram Showing Process of Transferring Gasoline from Large Tank Carried by Refueling Plane to Large Main Tanks of the "Question Mark" from Which the Fuel is Pumped to the Wing Tanks by Hand Pumps.

and accessible to mechanics, propellers being gear driven. By using a somewhat longer hose any possibility of contact between the two planes in formation could be eliminated, and by periodically adjusting the engines before they begin to lose power the big plane could be kept in full flight across the country. Such large planes would probably be accommodated only at special terminals of each coast and would make no intermediate stops.

How Fuel and Oil Were Transferred While in Flight.—During the refueling operations Captain Eaker would fly the Fokker in normal level flight while the refueling plane came up from the rear and took position about twenty feet above the "Question Mark" and a very little bit ahead, usually with the tail skid about over Major Spatz's head as shown at Fig. 95 C. Major Spatz completed the refueling connection by opening the trap door in the roof and standing on a raised platform with his head and shoulders out of the plane, then as the 1½-inch hose was lowered, he would grasp it and insert it into the funnel which fed the reserve tanks, thereafter holding the hose in place till the fueling operation was completed. Normally there was no perceptible variation in the distance between the two planes but on three occassions, the refueling plane pulled away and showered Major Spatz with gasoline before the operator above was able to stop the flow. The drawing at Fig. 95 D shows the internal arrangement of tanks in supply plane and the "Question Mark."

Oil was passed to the "Question Mark" through the trapdoor in the cabin roof, being received in five gallon cans and poured into a 60 gallon reserve tank in the main cabin. Penzoil triple extra heavy was the only engine oil used, and it was pumped to the engine tanks by means of a large hand operated wobble pump on the left forward side of the cabin.

Probably the most interesting part of the special equipment was that by which the Penzoil triple extra heavy in the engines and engine tanks was periodically drained and replaced by fresh oil; and the manner in which all rocker arms of all three engines were kept lubricated from the main cabin.

From the reserve supply of lubricating oil, copper tubes led to the three engine tanks, permitting these tanks to be filled at will. Valves were also provided at each tank by which they could be drained of old oil. On the nacelles these valves were located below and to the rear of the nacelle proper and were operated by torque rods connected to an indicating board within the cabin, one on the right side and one on the left, which told whether the valves were feeding oil to the tank, draining it out, or were in the closed position. Periodically the mechanic, Sergt. Roy Hooe, would turn the oil drain valve from inside the cabin, drain out the worn oil, close the oil drain valve, open the feed valve and pump the engine tank full with the wobble pump, thus keeping fresh oil in all engines throughout the flight.

For greasing the rocker arms a system of copper tube leads was installed, all of which terminated at a board in the forward part of the main cabin. There were three typical Alemite connections on the board, one connection for the pipe line leading to each engine. In order to grease the valve mechanism of any particular engine Sergeant Hooe simply connected an ordinary Alemite pressure gun to the proper terminal and shot Alemite

grease through the pipe line to the rocker arms of the desired engine. The pipe line to each engine was divided between the upper two cylinders and lead in series from rocker arm to rocker arm each way around the engine until the lines joined at the bottom.

Vacuum Fuel Feed.—One of the marked tendencies in automobile engineering has been the adoption of a vacuum fuel feed system to draw the gasoline from tanks placed lower than the carburetor instead of using either exhaust gas or air pressure to achieve this end. The device generally fitted is the Stewart vacuum feed tank which is clearly shown in section at Fig. 96. In this system the suction of a motor is employed to draw gasoline

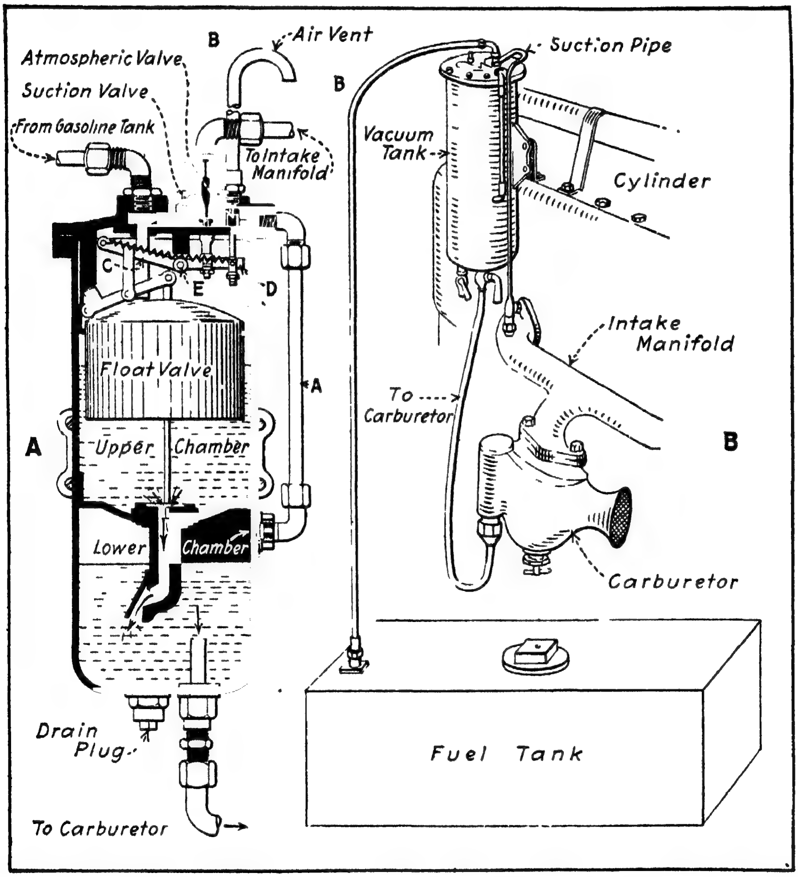


Fig. 96.—The Stewart Vacuum Fuel Feed Device Used on Numerous Motor Car Engines.

from the main fuel tank to the auxiliary tank incorporated in the device and from this tank the liquid flows to the carburetor. It is claimed that all the advantages of the pressure system are obtained with very little more complication than is found on the ordinary gravity feed. While the device has been widely applied to automotive engines used in motor cars, it has not been used in airplanes to any extent, because to have a device of proper

capacity would call for considerable weight and the electric or mechanical fuel pump systems or displacement by air pressure answer all requirements. The mechanism is all contained in the cylindrical tank shown, which may be mounted either on the front of the bulkhead or on the side of the engine as shown.

The tank is divided into two chambers, the upper one being the filling chamber and the lower one the emptying chamber. The former, which is at the top of the device, contains the float valve, as well as the pipes running to the main fuel container and to the intake manifold. The lower chamber is used to supply the carburetor with gasoline and is under atmospheric pressure at all times, so the flow of fuel from it is by means of gravity only. Since this chamber is located somewhat above the carburetor, there must always be free flow of fuel.

Atmospheric pressure is maintained by the pipes A and B, the latter opening into the air. In order that the fuel will be sucked from a main tank to the upper chamber, the suction valve must be opened and the atmospheric valve closed. Under these conditions the float is at the bottom and the suction at the intake manifold produces a vacuum in the tank which draws the gasoline from the main tank to the upper chamber. When the upper chamber is filled at the proper height the float rises to the top, this closing the suction valve and opening the atmospheric valve. As the suction is now cut off, the lower chamber is filled by gravity owing to there being atmospheric pressure in both upper and lower chambers. A flap valve is provided between the two chambers to prevent the gasoline in the lower one from being sucked back into the upper one. The atmospheric and suction valves are controlled by the levers C and D, both of which are pivoted at E, their outer ends being connected by two coil springs. It is seen that the arrangement of these two springs is such that the float must be held at the extremity of its movement, and that it cannot assume an intermediate position.

This intermittent action is required to insure that the upper part of the tank may be under atmospheric pressure part of the time for the gasoline to flow to the lower chamber. When the level of gasoline drops to a certain point, the float falls, thus opening the suction valve and closing the atmospheric valve. The suction of the motor then causes a flow of fuel from the main container. As soon as the level rises to the proper height the float returns to its upper position. It takes about two seconds for the chamber to become full enough to raise the float, as but .05 gallon is transferred at a time. The pipe running from the bottom of the lower chamber to the carburetor extends up a ways, so that there is but little chance of dirt or water being carried to the float chamber.

If the engine is allowed to stand long enough so that the tank becomes empty, it will be replenished after the motor has been cranked over four or five times with the throttle closed. The installation of the Stewart Vacuum-Gravity System is very simple. The suction pipe is tapped into the manifold at a point as near the cylinders as possible, while the fuel pipe is inserted into the gasoline tank and runs to the bottom of that member. There is a screen at the end of the fuel pipe to prevent any trouble due to deposits of sediment in the main container. As the fuel is sucked from the

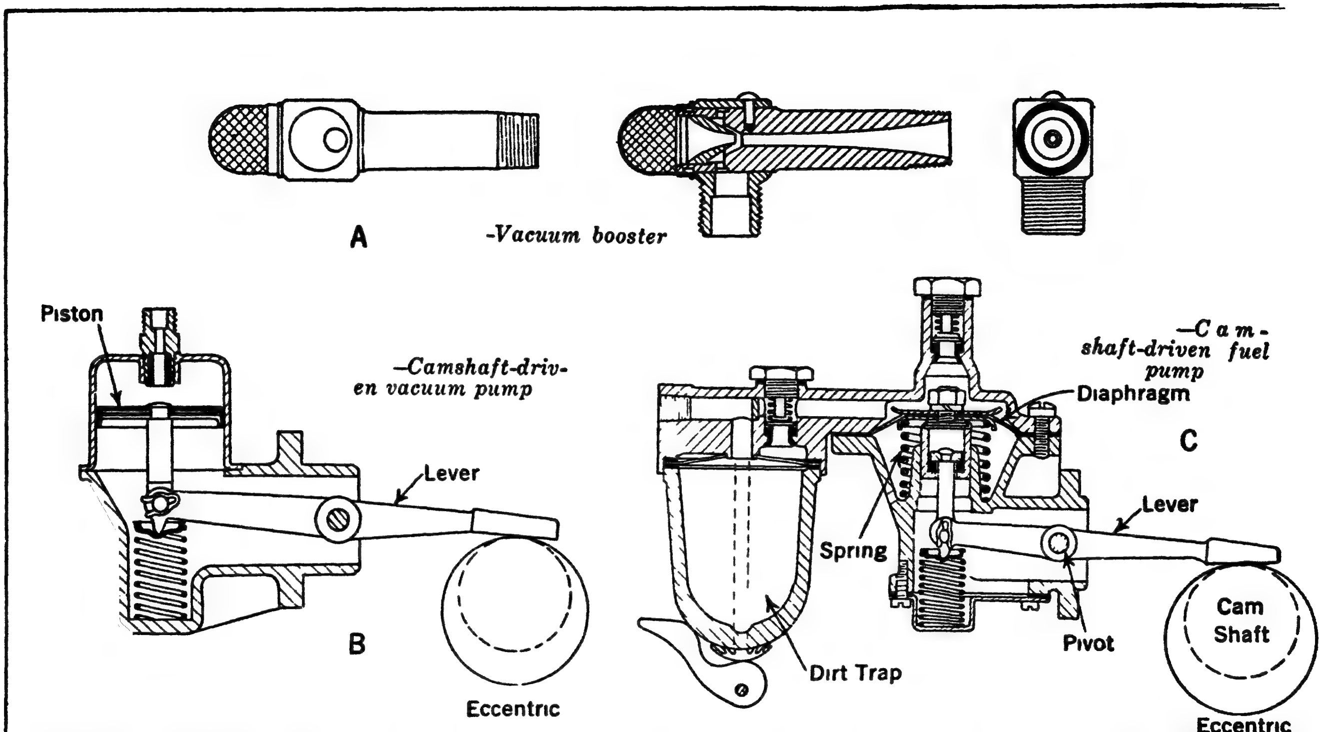


Fig. 97.—Recently Developed Devices for Fuel Feed. A—Vacuum Booster for Use with Stewart Vacuum Tank. B—Camshaft Driven Vacuum Pump to Supplement Manifold Suction when Vacuum Supply Tank is Used. C—Camshaft Driven Pump of Diaphragm Type Supplies Gasoline Directly from Main Tank to Carburetor.

gasoline tank, a small vent must be made in the tank filler cap so that the pressure in the main tank will always be that of the atmosphere.

There is some question as to the reliability of such a fuel feed system if installed in an airplane, when it is maneuvered or stunted. One of the difficulties that has prevented the airplane engine designer making use of this system is that aircraft engines are operated with large throttle openings, compared to automobile engines and the suction in the manifold would be lower than in an automobile, which is operated at less than maximum engine speed most of the time. The airplane engine must be delivering full power to attain maximum air speed and even at cruising speeds it is running at three-quarter full throttle so the suction effect is not enough to insure absolute fuel feed at all times.

Vacuum Booster.—Although it has been in use now for several months by a number of automobile manufacturers, the vacuum booster should also be included in the list of new developments. It is well known that in a number of modern engines, inlet manifold vacuums have fallen so low that vacuum tank operation is sometimes unsatisfactory, especially at the lower speeds with full throttle operation. To obviate this without resorting to mechanical or electrical means, the booster shown in Fig. 97 is offered. It is merely an injector admitting additional air to the manifold, the injector action increasing the vacuum at the tank, it is claimed, around 400 per cent at full throttle. Optionally with this is offered the camshaft-driven vacuum pump also shown at Fig. 97. It makes possible the use of a smaller vacuum tank. The function of the pump is to maintain a definite minimum vacuum for satisfactory operation. As long as the manifold vacuum is above this point its operation has no effect, it being closed off by a check valve between it and the manifold vacuum outlet connection.

Stewart Fuel Pump.—Another unit offered is the mechanically-operated fuel pump also shown at Fig. 97. This does not differ in principle from similar designs on the market, being driven from the camshaft by an eccentric. Its variable stroke for the diaphragm is obtained by a "piston" pulling on the diaphragm in proportion to the back pressure from the carburetor float chamber. A leather cushion for the piston is provided to absorb the shock.

The makers of the vacuum tank, realizing that there is a variety of automotive applications where the simple gravity-vacuum feed tank is not well adapted have designed other systems. The Stewart-Warner Speedometer Corporation have not only improved the vacuum tank systems of fuel feed but have developed other interesting types. These include:

1. A self-contained system incorporating in one unit a carburetor, a magnetic fuel pump and a pump regulator (Fig. 98).

2. A variation of this system having the carburetor and pump control in one unit and the pump itself located in the gas tank.

3. A further variation with all three units separated, the pump in the tank, the regulator on the dash, and capable of using any carburetor.

4. A standard system using a mechanically operated fuel pump which is operated from the camshaft as shown at Fig. 97.

5. An improved vacuum tank type of feed.

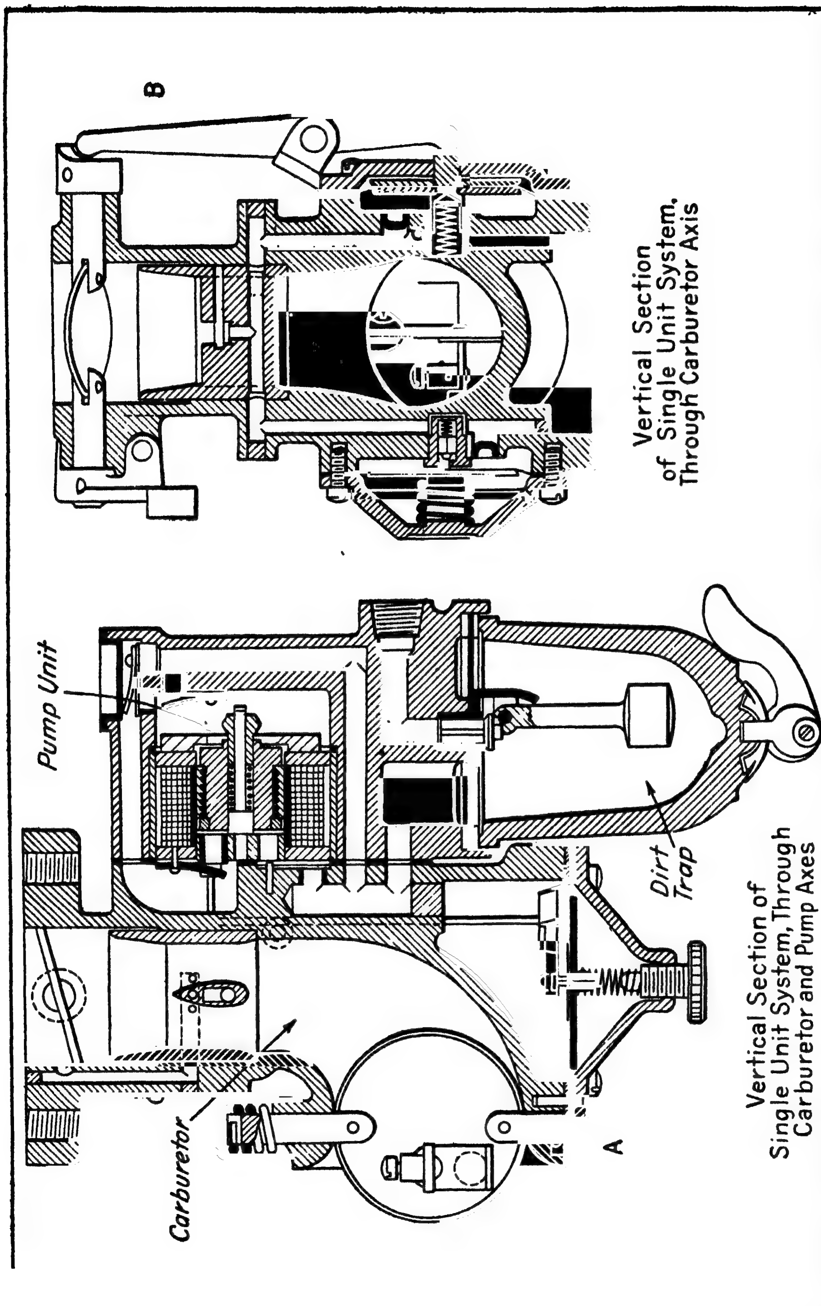


Fig. 98.—Stewart Carburetor at A, Combining Electrical Fuel Pump and Filtering Device in a Single Unit. Vertical Section of Single Unit System through the Carburetor Axis Shown at B.

6. The vacuum tank system supplemented by a vacuum booster mounted either on the intake manifold or on the tank itself.

7. A combination of vacuum tank and vacuum pump, the latter driven from the camshaft and effective only at low manifold vacuums to supplement tank action.

Each system has specific advantages. In principle both vacuum tank and mechanically operated pump systems are no longer experimental. The first three mentioned are unique, however, in that they are electrically operated. There have been some objections in the past to engine driven pumps on the ground that they cannot function until the engine has been started. The electrically-operated and controlled systems, on the other hand, can be and are made to function immediately on the turning on of the ignition switch. It is also claimed that the electric plunger pump will function better in cold weather than the diaphragm type. In addition, the electrically-operated and controlled pump is a true accessory, in that it does not require building into the engine.

Electrical Fuel Pump.—Taking the first of the three systems that use the electrical pump, that of the self-contained unit, its advantages are stated to be compactness, reduction of labor required for installation, and lower cost than for the mechanically operated pump system, especially when necessary engine changes are taken into account. The reason for using an electric pump in the self-contained unit rather than a restriction to bring the fuel to the carburetor is that the latter tends to choke the engine. A vacuum operated pump, moreover, would require too large a diaphragm for efficient operation with the low manifold inlet vacuums available at full throttle. The pump itself is of the magnetically-operated plunger type and has a capacity of seventeen gallons per hour in its present form. The plunger is composed of magnet iron fitted with bakelite rings, alternate rings fitting the plunger shaft and the brass sleeve in which the plunger rides, thus forming a self-adjusting seal. By this means the necessity of grinding or fitting any metal parts to close limits is eliminated, reducing the cost. To assemble the rings on the plunger, a metal packing ring is pressed on. Reference to Fig. 98 A will make the design clearer. Pressed onto a pin pressed into the plunger at the outer end is a hardened cam whose function is to trip the pump circuit. A hardened roller rides on the cam shown in Fig. 98 A. In the normal position this circuit is closed. When the diaphragm in the carburetor closes the main circuit, which is in series with that of the pump, the magnet core is energized and draws the plunger in against spring tension. The cam finally trips the circuit open and the plunger returns to normal position again, closing the pump circuit. The circuit breaker mechanism is mounted on a bakelite disc for insulation. The pump is a double-acting type. All pump valves are of the flapper, spring-seated type, very light in operation. At the inner end of the pump holes are drilled through the plunger stop. The inlet valve at the bottom is mounted on the pump mounting carburetor flange, and the outlet on the pump casing. Operation of the other side of the pump is quite similar, both the inlet and the outlet flapper valves being located on the pump housing. As will be noted from sectional views, the contact points for the elec-

trical circuit are located in compartments containing fuel. This might lead one to expect a considerable fire hazard. Stewart-Warner engineers claim, however, that they have been unable so far to fire any combustible mixture by means of these breaker points. The current required is very light in amperage and is taken from an ordinary six volt storage battery.

Coincident with the development of the self-contained unit a new carburetor was developed by Stewart-Warner, which has a fixed diaphragm in place of the usual float valve, the diaphragm being controlled by a column of fuel above it. With the built-in electric pump this diaphragm is made use of to control the action of the electric pump by making it open and close an electric circuit. It is claimed that with this design much closer regulation of the fuel level can be obtained.

Incidentally, the suction-operated diaphragm also operates the economizer through a small valve. The carburetor is also equipped with a fourth "jet" to provide a manual control if it is desired to enrich the mixture throughout the range of air speeds, as in warming up. It is brought into action by pulling out the choke button and remains in action after release of the choke button and closing of the choke valve, until a secondary dash control button, pulled out by the choke, is pushed in.

The second and third systems do not differ in operation from the single unit fuel feed. In both of these the pump is located in the gas tank at the rear. The design of the pump is a separate unit for such installation. The only difference between the two systems is that the first retains the electric pump control on the Stewart carburetor, while the latter permits the use of any carburetor through separate location of the pump control, which in turn is controlled by the carburetor float mechanism as with mechanically-operated fuel pumps.

One of the features of the improved tank is the elimination of all springs, which should make its operation more reliable. The check valve to the outer chamber is extremely simple. It consists merely of a small fabric disc fastened on by a spring clip, the seat being on the metal rather than on the valve. Operation of the tank is by means of a float which has a brass plate of 0.050 in. thickness attached to it at the bottom for weighting, the float guide having been eliminated. The tank is furnished either with or without integral fuel strainer. It is fastened on by means of a spring clip provided with an eccentric to lock it in place. This strainer, if furnished, is equipped with a shut-off valve so that the removal of the bulb for cleaning will divert the fuel flow directly to the carburetor.

Barlow Fuel Pump.—A new fuel pump developed by Lester P. Barlow, is being offered by the McCord Radiator & Mfg. Co. for automotive and aircraft use. The new pump is designed so that it can be driven from nearly any part of the engine, including a generator shaft, camshaft, or fan shaft, and embodies a roller principle in design. In construction it is composed mainly of a flanged casting, into which is inserted a smaller round casting carrying the intake, outlet and balancing ports. The pump shaft passes through the center of both castings and has a disc at the pump end in which are four grooves giving it the shape of a Maltese Cross. Four steel rollers ride in these grooves as shown at Fig. 99. The pump shaft and

disc are eccentrically mounted in the pump casting, so that the rollers under centrifugal action ride away and toward the axis of rotation during a revolution, the clearance at the bottom between the pump disc and the hardened steel ring against which the rollers ride providing the space through which the fuel is pumped by the rollers.

The disassembled view at bottom of Fig. 99 shows the port casting. The three outer ports on one side and the annular port on the same side serve as intake parts and those on the other side for the outlet. The reason for using the inner annular ports is to provide a balancing action for the rollers, as well as to aid the self-priming action of the pump. There is also a slight clearance between the rollers and the grooves in which they ride so

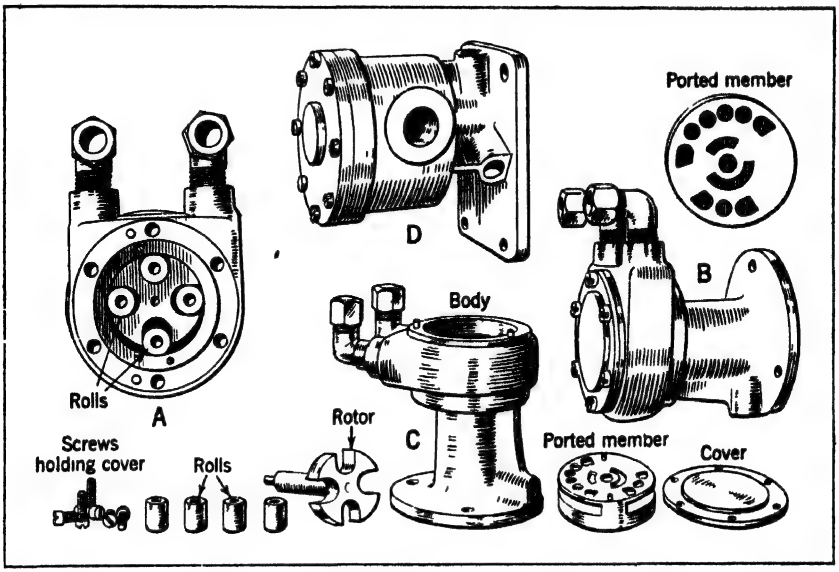


Fig. 99.—The Barlow Fuel Pump Incorporates a Novel Roller Principle. A—Front View of Automobile Type Pump. B—Side View of Automobile Type Pump. C—Parts of Pump. D—Aircraft Fuel Pump.

as to complete the balancing action, and to provide some additional pumping action by radial flow of the liquid to the outer ports, as well as through the annular central outlet ports. Two variations of this pump are being offered, one for automotive and one for aircraft engine uses, the latter incorporating a by-pass arrangement and providing somewhat greater flow of liquid. While announced as a fuel pump it is also adaptable for other purposes, such as the providing of supply water for steam-cooling systems. The pump is particularly efficient as regards suction due to the self-priming action, and the centrifugal pressure of the rollers against the hardened and ground ring, aided by the weight of the rollers themselves. It will operate at very low speeds, and can be made to operate by merely turning the shaft with the fingers.

Biflex Fuel Pump.—A new type of fuel pump using a Sylphon metal bellows, used for some time in various thermostatic and mechanical devices is shown at Fig. 100. This new pump was described and illustrated in *Automotive Industries* and is suited for all automotive applications. In this pump fuel is received inside the bellows on the intake stroke or when the bellows are expanded, and is forced to the carburetor on the exhaust stroke or when the bellows are contracted. Two models of pump are offered differing only in the means used to operate the bellows. In one, shown in the accompanying illustration, a rocker arm operated by an eccentric is employed to expand and contract the bellows while the other model employs a push rod instead of a rocker arm. When the movement of the

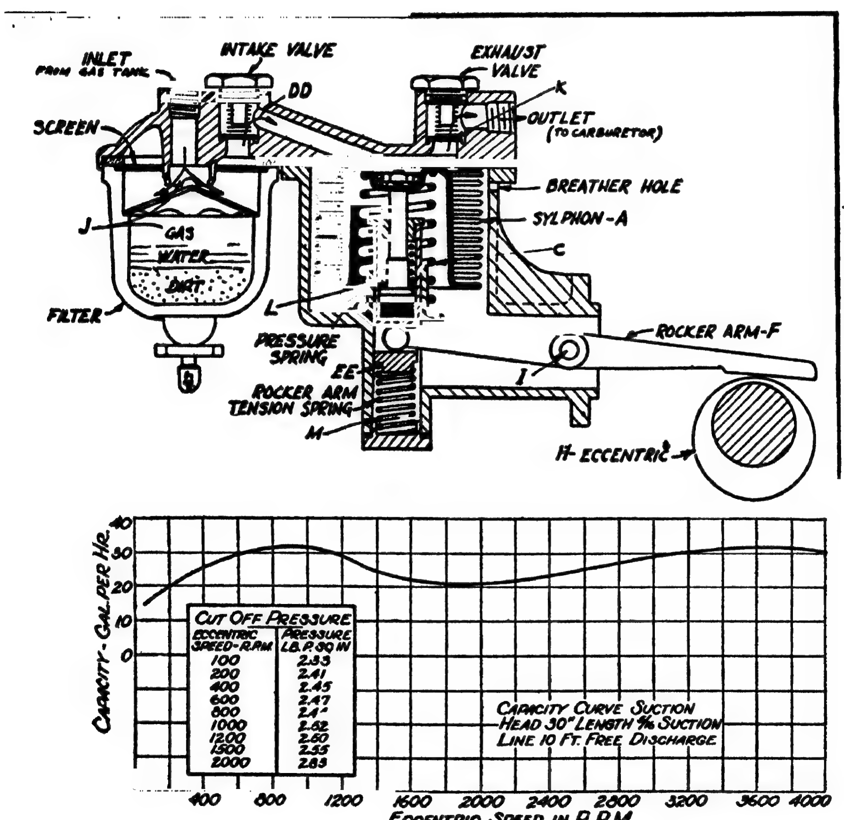
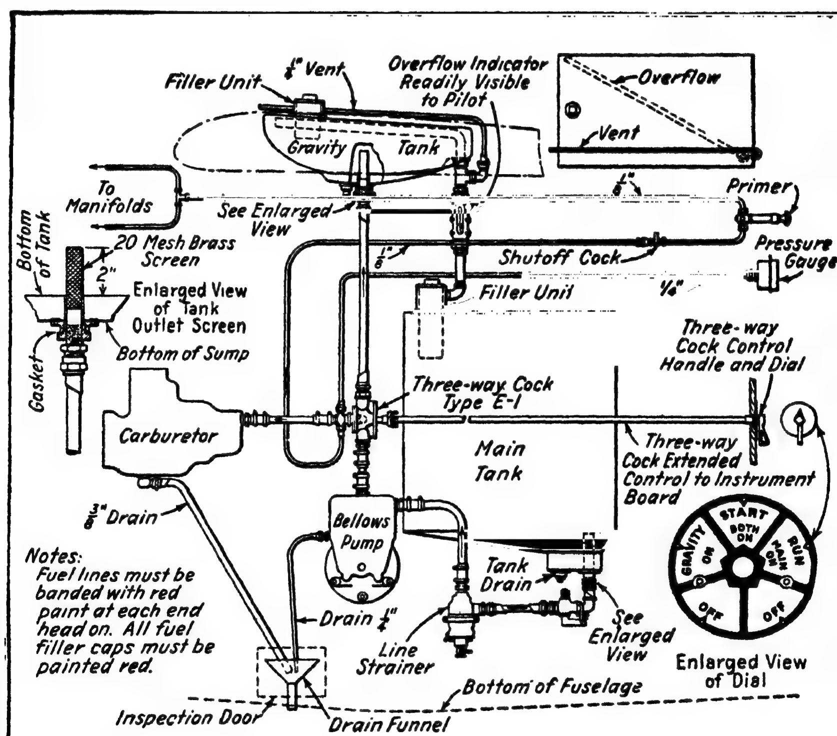


Fig. 100.—Diagram Showing Construction of Biflex Fuel Pump and Curve Showing Operating Test Data.

rocker arm reciprocates the plunger EE the bellows is expanded, causing a vacuum in the intake passage D and causing fuel to flow from the main tank through the filter J and into the bellows. Further operation of the eccentric compresses the bellows, forcing the fuel therein through the outlet valve to the carburetor.

After a few repetitions of this movement, when starting an engine, the pumping chamber and carburetor bowl are filled with fuel and the pressure spring C and the resilient action of the Sylphon bellows produce a predetermined pressure upon the fuel supplied to the carburetor. As the pressure increases the movement imparted to the bellows head decreases under the action of the cushion spring L and the movement will finally be adjusted to the proper operating conditions for which the device was designed. The spring C and the resilience of the bellows determine the pressure under which fuel is delivered, and these are usually selected so that cut-off or dead-end pressures are not more than about 2½ pounds per square inch. The bellows is made of metal, is unaffected by atmospheric conditions and operates satisfactorily in warm, cold, dry or wet weather. It resists ordinary corrosion and is unaffected by gasoline. The pump is self-priming under engine starter speeds.



Air Service Typical Fuel System

Fig. 101.—Diagram Showing Parts of U. S. A. Air Service Typical Fuel System.

Air Service Typical Fuel System.—The illustration at Fig. 101 shows a diagram of a typical fuel system for airplanes in which the gasoline is pumped by an engine driven bellows pump and supplied to an emergency gravity tank in the center section. When more than one engine is used, as in a bimotor or trimotor plane, a separate wing tank is usually supplied for each engine, with arrangements for filling all three from the main tank

or tanks in the fuselage. The main fuel supply passes through a line strainer shortly after leaving the tank and is coupled to the intake of the bellows pump. From the outlet of the bellows pump it passes through a three-way cock to the bottom of the gravity tank. A dial on the bulkhead indicates the proper placing of the valve control handle for various fuel feed combinations. In starting, both tanks are connected to the carburetor. In running the pump delivers fuel directly to the carburetor because the emergency tank is shut off. If any trouble develops in the main fuel supply, the valve handle is placed in the position that the dial indicates is necessary to have the gravity feed tank connected to the carburetor. An overflow indicator in the overflow pipe leading from gravity tank to the filler fitting or unit of the main tank is readily visible to the pilot and shows when the gravity tank is full after starting at which time the valve handle should be placed at "Run" position and fuel pumped direct from main tank to carburetor. Two small leads are taken off from the feed line just forward of the three-way cock and one is used to supply the fuel pressure gauge, the other a hand-operated primer unit. A drain pipe from the carburetor and one from the pump communicate with a drain funnel at the botton of the fuselage. The general arrangement, in view of the explanation given can be easily followed and various lines traced by studying the diagram.

Direct Fuel System.—One of the latest forms of combined fuel supply and carburetion system is known as the direct fuel system and has been described by F. G. Whittington in the *S. A. E. Journal* for December, 1928. It is stated that this system is well adapted to airplane service and that it was designed with this application in view. Tests have shown remarkable performance and well known automotive engineering authorities have endorsed it.

The layout calls for elimination of the conventional type of carburetor, the vacuum tank or the fuel pump. It is designed with the idea of supplying an approximately dry mixture of gas and air in the correct proportions to the inlet manifold without any intermediate storage of fuel from the time this fuel leaves the main supply tank. The system is based on the theory that thorough vaporization of the fuel is the best method of carburetion, as thorough vaporization is regarded as a requisite for complete combustion. With the increasing engine speeds, the fuel-vaporization problem becomes very important. The atomization action of carburetors has been improved but, in addition, it also is necessary to provide interchange of heat between exhaust and inlet manifolds. Improvements in manifolds and hot-spot designs have greatly improved conditions, but room still remains for further development. With the present tendency toward increasing the number of cylinders per powerplant unit, manifolding is becoming increasingly difficult because of the compromise between volumetric efficiency and distribution.

The direct fuel system meets the compromise between volumetric efficiency and distribution by providing the gases for engine operation in a highly vaporized state and permits large manifolding, which, in itself, is a construction favorable to volumetric efficiency. An additional favorable feature of this system is the provision for the passing of part of the charge through more intense heat than is usually found in the present methods of

hot-spot application. A brief description of the complete installation is as follows:

In the main fuel supply tank a down pipe of approximately ⅝-inch diameter is installed, which carries air to the lowest part of the tank. There it forms a junction with the properly proportioned jetting means and produces a mixture that is carried through a conduit to a heating element mounted directly in the exhaust manifold. At this point, where very high temperatures are encountered, the heavier part of the mixture is acted upon. The lighter ends pass on through and, at the point of vaporization, the heavier ends are passed on to the air-mixing chamber. At this point the gases come entirely under throttle control under usual carbureting conditions. In this installation the following conditions exist:

(1) Vaporization is assured by the introduction of warm air at the rear jets, the long feed-tube that is under less than atmospheric pressure, and the efficient design of the heating element.
(2) Low idling speeds as well as varying load and speed conditions are secured by proper proportioning of jets.
(3) Condensed portions of fuel in the conduit provide a satisfactory accelerating supply.
(4) Faster warming-up is possible, and easy starting is accomplished.
(5) High volumetric efficiency is maintained by the admission of cool air to the vaporized fuel mixture in the approximate proportion of four to one.
(6) The installation is comparatively simple, various storage points for fuel, such as the vacuum tank and the carburetor float bowl, being eliminated.
(7) Thorough vaporization assures smoothness in operation, better economy, and increased horsepower.

Regardless of how radical the direct fuel system may seem, its performance is and can be considered as another reliable means of fuel supply.

Fuel Supply Systems Summarized.—The following covers the various means of fuel supply:

(1) A standard carburetor and a vacuum tank that takes its fuel from the supply tank and operates on available vacuum at the inlet manifold, used principally on automobiles.
(2) A standard carburetor, a vacuum tank and a booster attachment to assure a supply of fuel at low inlet-manifold vacuum, also employed on motor vehicles.
(3) A standard carburetor, a vacuum tank and a trap valve to assure a vacuum sufficient to supply fuel under all working conditions. Successful operation on all engines requires a late inlet-valve timing.
(4) A standard carburetor, a vacuum tank and a vacuum pump. The use of the pump gives high vacuum at all times, as it coordinates with the inlet-manifold vacuum.
(5) A standard carburetor and a mechanically driven fuel pump of diaphragm, plunger, eccentric vane, sylphon bellows or any other type that pumps fuel directly from the main supply tank and delivers it to the carburetor under safe working pressure under variable

flow-requirements. This is a popular system for aircraft applications.

(6) A standard carburetor and an electromagnetic fuel pump of plunger type controlled by a separate circuit-control unit. The pump is mounted directly in the main supply tank and the control is mounted convenient to the operator.

(7) A combined carburetor and circuit-control unit and an electromagnetic fuel pump of plunger type. The carburetor and control member are built as a single unit and the pump is mounted in the main fuel supply tank.

(8) A self-contained unit incorporating a carburetor, a circuit-control unit, and an electromagnetic fuel pump of plunger type. (Stewart-Warner System.)

(9) The direct fuel system using no conventional form of carburetor, no vacuum tank and no fuel pump. It takes fuel directly from the main fuel supply tank by special jetting means, vaporizes it, and adds cool air to approximate a dry-gas mixture at the engine. (Stewart-Warner System.)

(10) A diaphragm-type carburetor and a vacuum tank or a fuel pump of any type. The carburetor is of a design that incorporates many improvements and assures in most cases a marked increase in engine output.

(11) A mechanical fuel pump delivering the fuel to a gravity tank from which it flows to the carburetor.

(12) Obsolete systems in which fuel is displaced from the main container by air or exhaust gas pressure and made to flow to the carburetor. Surplus air released through an excess pressure relief valve.

Principles of Carburetion Outlined.—The process of carburetion is combining the volatile vapors which evaporate from the hydrocarbon liquids with certain proportions of air to form an inflammable gas. The quantities of air needed vary with different liquids and some mixtures burn quicker than do other combinations of air and vapor. Combustion is simply burning and it may be rapid, moderate or slow. Mixtures of gasoline and air burn quickly, in fact the combustion is so rapid that it is almost instantaneous and we obtain what is commonly termed an "explosion." Therefore, the explosion of gas in the automobile engine cylinder which produces the power is really a combination of chemical elements which produce heat and an increase in the volume of the gas because of the increase of temperature as the reader has learned from the chapters on thermodynamics.

If the gasoline mixture is not properly proportioned the rate of burning will vary, and if the mixture is either too rich or too weak the power of the explosion is reduced and the amount of power applied to the piston is decreased proportionately. In determining the proper proportions of gasoline and air, one must take the chemical composition of the fuel used into account. The ordinary liquid used for fuel is said to contain about eighty-four per cent carbon and sixteen per cent hydrogen. Air is composed of oxygen and nitrogen and the former has a great affinity, or combining power, with the two constituents of hydrocarbon liquids. Therefore, what we call an explosion is merely an indication that oxygen in the air has com-

bined with the carbon and hydrogen of the gasoline so rapidly that heat is generated.

Air Needed to Burn Gasoline.—In figuring the proper volume of air to mix with a given quantity of fuel, one takes into account the fact that one pound of hydrogen requires eight pounds of oxygen to burn it, and one pound of carbon needs two and one-third pounds of oxygen to insure its combustion. Air is composed of one part of oxygen to three and one-half portions of nitrogen by weight. Therefore for each pound of oxygen one needs to burn hydrogen or carbon four and one-half pounds of air must be allowed. To insure combustion of one pound of gasoline which is composed of hydrogen and carbon we must furnish about ten pounds of air to burn the carbon and about six pounds of air to insure combustion of hydrogen, the other component of gasoline. This means that to burn one pound of gasoline one must provide about sixteen pounds of air.

While one does not usually consider air as having much weight, at a temperature of 62 deg. F. about fourteen cubic feet of air will weigh a pound, and to burn a pound of gasoline one would require about 200 cubic feet of air. This amount will provide for combustion theoretically, but it is common practice to allow twice this amount because the element nitrogen, which is the main constituent of air, is an inert gas and instead of aiding combustion it acts as a deterrent of burning. In order to be explosive, gasoline vapor must be combined with definite quantities of air. Mixtures that are rich in gasoline ignite quicker than those which have more air, but these are only suitable when starting or when running slowly, as a rich mixture ignites easier than a weak mixture having an excess of air. The richer mixture of gasoline and air not only burns quicker but produces the most heat and if combustion is complete most effective pressure in pounds per square inch of piston top area.

The amount of compression of the charge before ignition also has material bearing on the force of the explosion as the writer and other authorities quoted have pointed out in a preceding chapter. The higher the degree of compression the greater the force exerted by the rapid combustion of the gas. It may be stated that as a general thing the maximum explosive pressure is somewhat more than four times the compression pressure prior to ignition. A charge compressed to 60 pounds will have a maximum of approximately 240 pounds; compacted to 80 pounds it will produce a pressure of over 300 pounds on each square inch of piston area at the beginning of the power stroke. Mixtures varying from one part of gasoline vapor to four of air to others having one part of gasoline vapor to seventeen of air can be ignited.

What a Carburetor Should Do.—While it is apparent that the chief function of a carbureting device is to mix hydrocarbon vapors with air to secure mixtures that will burn, there are a number of factors which must be considered before describing the principles of vaporizing devices. Almost any device which permits a current of air to pass over or through a volatile inflammable liquid will produce a gas which will explode when compressed and ignited in the motor cylinder. Modern carburetors are not only called upon to supply certain quantities of gas, but these must deliver a mixture to the cylinders that is accurately proportioned and which

will be of proper composition at all engine speeds.

Flexible control of the engine in an automobile or airplane is sought by varying the engine speed by regulating the supply of gas to the cylinders. The powerplant should run from its lowest to its highest speed without any irregularity in torque, i.e., the acceleration should be gradual rather than spasmodic. As the degree of compression will vary in value with the amount of throttle opening, the conditions necessary to obtain maximum power differ with varying engine speeds. When the throttle is barely opened the engine speed is low and the gas must be richer in fuel than when the throttle is wide open and the engine speed high. When an engine is turning over slowly with throttle nearly closed the compression has low value and the conditions are not so favorable to rapid combustion as when the compression is high. At high engine speeds the gas velocity through the intake piping is higher than at low speeds, and regular engine action is not so apt to be disturbed by condensation of liquid fuel in the manifold due to excessively rich mixture or a superabundance of liquid in the stream of carbureted air.

For a long time carburetor engineers held to the theory that a carburetor should furnish a mixture of constant strength throughout its range of operation, but this has now been disproved. At the laboratories of the Research Association of British Motor and Allied Manufacturers an investigation on the effect of changes in fuel mixture ratio on the maximum power output and the thermal efficiency has been made on a single cylinder 3.5 hp. engine with a compression ratio of 3.86:1. The results obtained were generally in agreement with those obtained by Professors Berry and Kegerreis in this country. That is, the maximum power is obtained from an engine if the fuel charge contains substantially twenty per cent more than the theoretically required amount of gasoline, while the mixture ratio giving the best economy varies with the load, from about 17.25:1 at full load to 12.8:1 at low load. The carburetion problem is concerned with more than the mere proportioning of the mixture. For rapid and efficient combustion thorough intermixture of the reactive molecules is required, which can be effected only by complete vaporization of the fuel.

QUESTIONS FOR REVIEW

1. What is the simplest method of fuel supply to carburetor?
2. Name possible locations of fuel tanks in airplane structure.
3. What are the disadvantages of air pressure systems?
4. What are the advantages of re-fueling in flight?
5. Describe action of vacuum fuel feed.
6. What is a vacuum booster and why is it used?
7. How does the Barlow pump work?
8. What is the principle of operation of the Biflex pump?
9. Describe typical U. S. Air Service Fuel system.
10. What should a carburetor do?

CHAPTER IX

DEVELOPMENT OF FLOAT FEED CARBURETOR

Early Vaporizer Forms—Marine Mixing Valve—Development of Float Feed Carburetor—Maybach's Early Design—Early Phoenix-Daimler Design—Concentric Float and Jet Type—Schebler Carburetor—The Claudel (French) Carburetor—Metering Pin Carburetor—Multiple Nozzle Vaporizers—Priming Necessary to Start Cold Engines—Function of Accelerating Wells—Ball and Ball Two-Stage Carburetor—Master Multiple Jet Carburetor—Notes on Adjustment of Simple Carburetors—Effect of Altitude Changes—Compound Nozzle Zenith Carburetor—Function of Compensator—Zenith Liberty Type—Zenith with Venturi Delivery Nozzle.

Early Vaporizer Forms.—The very early types of carbureting devices were very crude and cumbersome, and the mixture of gasoline vapor and air was accomplished in three ways so that before modern carburetors are described, it may be well to discuss briefly methods that have been employed. The air stream was passed over the surface of the liquid itself, through loosely placed absorbent material saturated with liquid, or directly through the fuel. The first type was known as the surface carburetor and is now practically obsolete. The second form was called the "wick" carburetor because the air stream was passed over or through saturated wicking. The third form was known as a "bubbling" carburetor. While these primitive forms gave fairly good results with the early slow-speed engines and the high grade, or very volàtile, gasoline which was first used for fuel, they would be entirely unsuitable for present forms of engines because they would not carburate the lower grades of gasoline which are used today, and would not supply the modern high-speed engines with gas of the proper consistency fast enough, even if they did not have to use very volatile gasoline. The form of carburetor used at the present time operates on a different principle. These devices are known as "spraying carburetors." The fuel is reduced to a spray by the suction effect of the engine piston and the entering air stream drawing it through a fine opening or a series of holes.

The advantage of this construction is that a more thorough amalgamation of the gasoline and air particles is obtained. With the earlier types previously considered the air would combine with only the more volatile elements, leaving the heavier constituents in the tank. As the fuel became stale it was difficult to vaporize it, and it had to be drained off and fresh fuel provided before the proper mixture would be produced. It will be evident that when the fuel is sprayed into the air stream, all the fuel will be used up and the heavier portions of the gasoline will be taken into the cylinder and vaporized just as well as the more volatile vapors.

Marine Mixing Valve.—The simplest form of spray carburetor is that shown at Fig. 102. In this the gasoline opening through which the fuel is sprayed into the entering air stream is closed by the spring-controlled mushroom valve which regulates the main air opening as well. When the engine draws in a charge of air it unseats the valve and at the same time the air flowing around it is saturated with gasoline particles through the

gasoline opening. The mixture thus formed goes to the engine through the mixture passage. Two methods of varying the fuel proportions are provided. One of these consists of a needle valve to regulate the amount of gasoline, the other is a knurled screw which controls the amount of air by limiting the lift of the jump valve. This mixing valve is employed only on marine engines at the present time as it has disadvantages, owing to having a rapidly moving valve the action of which is erratic at high speeds, that make its use on aviation engines impractical.

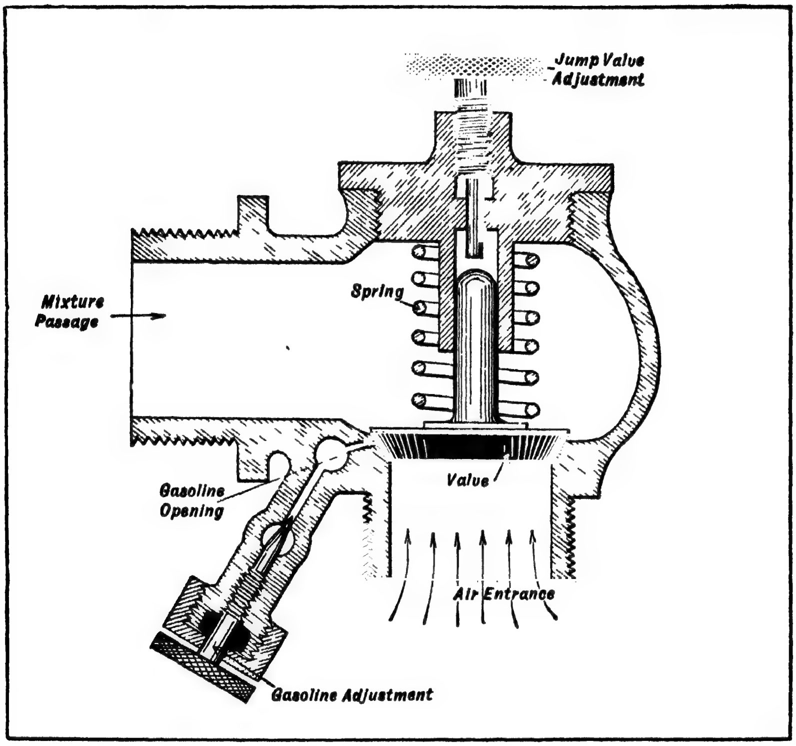


Fig. 102.—Marine Type Mixing Valve by which Gasoline is Sprayed into Air Stream Through a Small Opening in Air Valve Seat is Simplest Carburetor Form, but is not Suited for Aircraft Engines.

Development of Float Feed Carburetor.—The modern form of spraying carburetor is provided with two chambers, one a mixing chamber through which the air stream passes and mixes with a gasoline spray, the other a float chamber in which a constant level of fuel is maintained by simple mechanism. A jet or standpipe is used in the mixing chamber to spray the fuel through and the object of the float is to maintain the fuel level to such a point that it will not overflow the jet when the motor is not drawing in a charge of gas. With the simple forms of generator valve in which the gasoline opening is controlled by the air valve, a leak anywhere in either valve or valve seat will allow the gasoline to flow continuously whether the engine is drawing in a charge or not. The liquid fuel collects around the air opening, and when the engine inspires a charge it is sat-

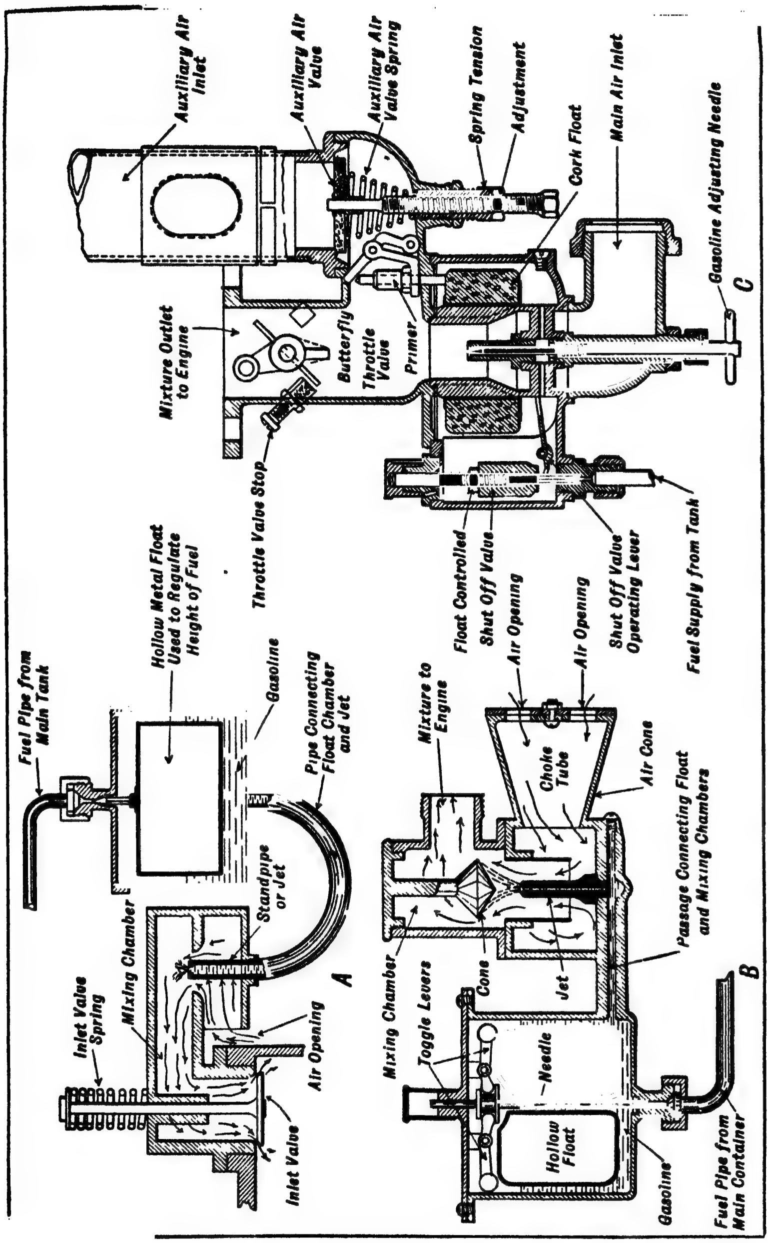


Fig. 103.—Diagram Tracing Evolution of the Simple Spray Carburetor. A—An Early Form Evolved by Maybach. B—Phoenix-Daimler Modification of Maybach's Principle. C—An Early Concentric Float Automatic Compensation Automobile Type Carburetor. These Types now are Obsolete and are Shown so Reader Can Compare them with Modern Types.

urated with gasoline globules and is excessively rich. With a float feed construction, which maintains a constant level of gasoline at the right height in the standpipe, liquid fuel will only be supplied when drawn out of the jet by the suction effect of the entering air stream.

Maybach's Early Design.—The first form of spraying carburetor ever applied successfully was evolved by Maybach for use on one of the earliest Daimler engines. The general principles of operation of this pioneer

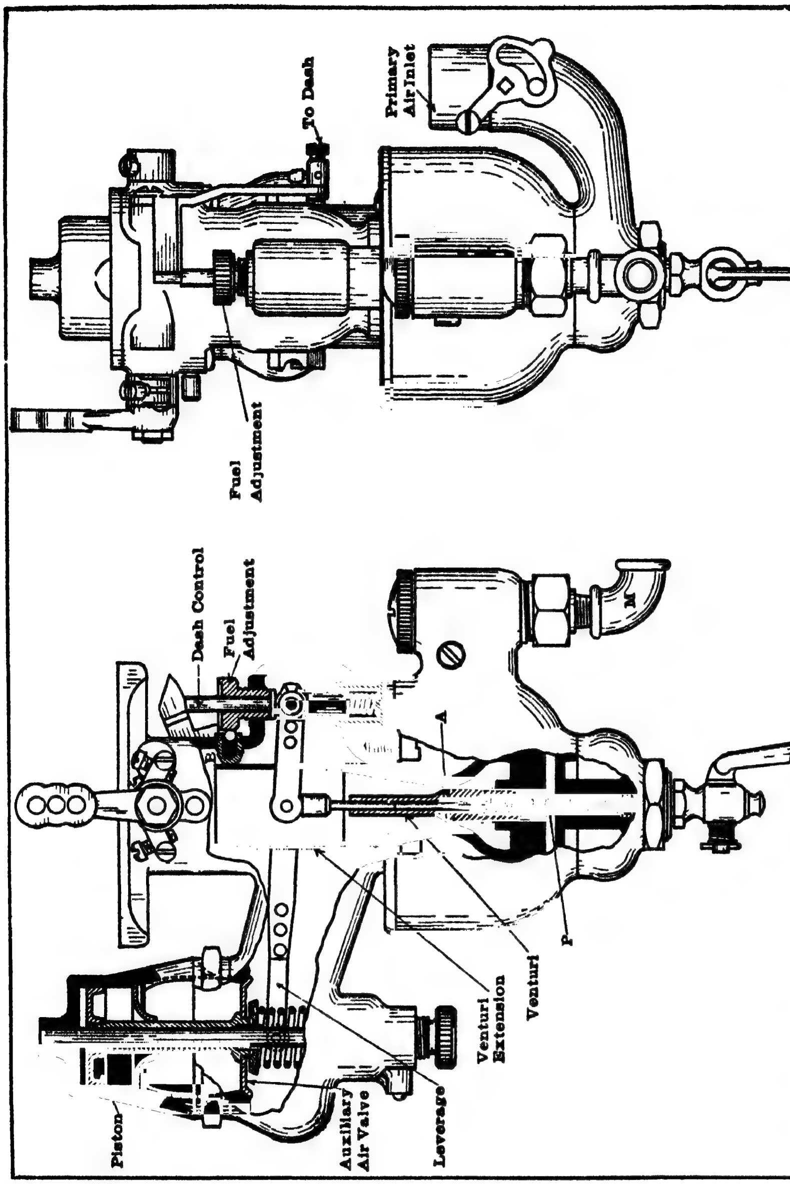


Fig. 104.—Early Model Schebler Carburetor with Metering Valve and Venturi. Note Mechanical Connection Between Auxiliary Air Valve and Fuel Regulating Needle. This Carburetor was used on Some Pre-War Engines.

float feed carburetor are shown at Fig. 103 A. The mixing chamber and valve chamber were one and the standpipe or jet protruded into the mixing chamber. It was connected to the float compartment by a pipe. The fuel from the tank entered the top of the float compartment and the opening was closed by a needle valve carried on top of a hollow metal float. When the level of gasoline in the float chamber was lowered the float would fall and the needle valve uncover the opening. This would permit the gasoline from the tank to flow into the float chamber, and as the chamber filled the float

would rise until the proper level had been reached, under which conditions the float would shut off the gasoline opening. On every suction stroke of the engine the inlet valve, which was an automatic type, would leave its seat and a stream of air would be drawn through the air opening and around the standpipe or jet. This would cause the gasoline to spray out of the tube and mix with the entering air stream.

Early Phœnix-Daimler Device.—The form shown at B was a modification of Maybach's simple device and was first used on the Phœnix-Daimler engines. Several improvements are noted in this device. First, the carburetor was made one unit by casting the float and mixing chambers together instead of making them separate and joining them by a pipe, as shown at A. The float construction was improved and the gasoline shut-off valve was operated through leverage instead of being directly fastened to the float. The spray nozzle was surrounded by a choke tube which concentrated the air stream around it and made for more rapid air flow at low engine speeds. A conical piece was placed over the jet to break up the entering spray into a mist and insure a more intimate mixture of air and gasoline. The air opening was provided with an air cone which had a shutter controlling the opening so that the amount of air entering could be regulated and thus vary the mixture proportions within certain limits.

Concentric Float and Jet Type.—The form shown at B has been further improved, and the type shown at C is representative of early automobile engine single jet carburetor practice. In this the float chamber and mixing chamber are concentric. A balanced float mechanism which insures steadiness of feed is used, the gasoline jet or standpipe is provided with a needle valve to vary the amount of gasoline supplied the mixture and two air openings are provided. The main air port is at the bottom of the vaporizer, while an auxiliary air inlet is provided at the side of the mixing chamber. There are two methods of controlling the mixture proportions in this form of carburetor. One may regulate the gasoline needle or adjust the anxiliary air valve. Such carburetors are seldom used with late designs of engines as compound compensating jet forms are most popular in automotive applications, especially on aircraft engines.

Schebler Carburetor.—A Schebler carburetor, which has been used on some early airplane engines, is shown in Fig. 104. It will be noticed that a metering pin or needle valve opens the jet when the air valve opens. The long arm of a leverage is connected to the air valve, while the short arm is connected to the needle, the reduction in leverage being such that the needle valve is made to travel much less than the air valve. For setting the amount of fuel passed or the size of the jet orifice when running with the air valve closed, there is a screw which raises or lowers the fulcrum of the lever and there is also a dash control having the same effect by pushing down the fulcrum against a small spring. A long extension is given to the venturi tube which is very narrow around the jet orifices, which are horizontal and shown at A in the drawing. Fuel enters the float chamber through the union M, and the spring P holds the metering pin upward against the restraining action of the lever. The air valve may be set by an easily adjustable knurled screw shown in the drawing, and

fluttering of the valve is prevented by the piston dash pot carried in a chamber above the valve into which the valve stem projects. The primary air enters beneath the jet passage and there is a small throttle in the intake to increase the speed of air flow for starting purposes. The carburetor is adapted for the use of a hot-air connection to the stove around the exhaust pipe and it is recommended that such a fitting be supplied. The lever which controls the supply of air through the primary air intake is so arranged that if desired it can be connected with a linkage on the bulkhead or control column by means of a flexible wire.

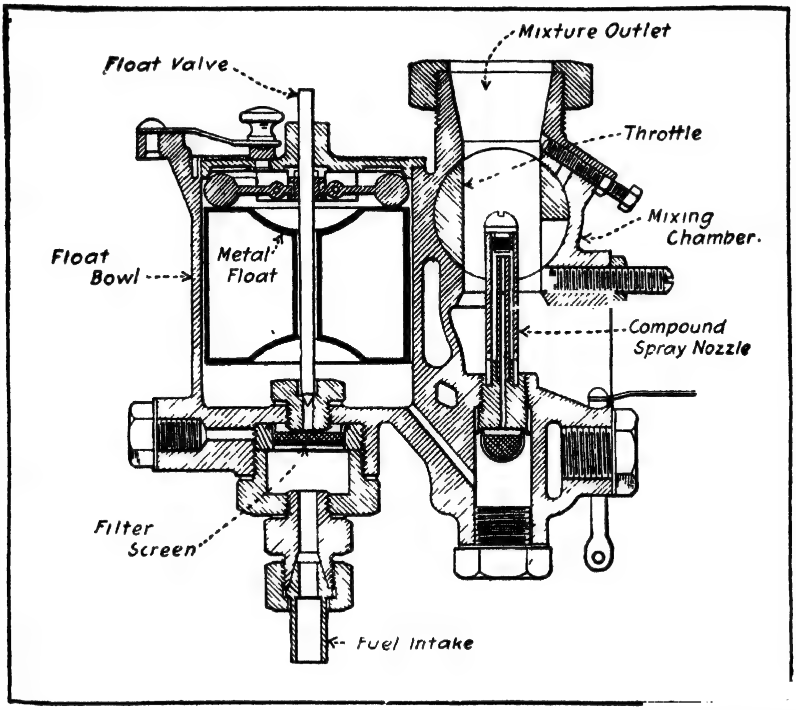


Fig. 105.—The Claudel Carburetor of Early Design was One of the First to be Applied to Aviation Engines.

The Claudel (French) Carburetor.—This carburetor is of extremely simple construction, because it has no supplementary or auxiliary air valve and no moving parts except the throttle controlling the gas flow. The construction is clearly shown in Fig. 105. The spray jet is concentric with a surrounding sleeve or tube in which there are two series of small orifices, one at the top and the other near the bottom. The former are about level with the spray jet opening. The sleeve surrounding the nozzle is closed at the top. The air, passing the upper holes in the sleeve, produces a vacuum in the sleeve, thereby drawing air in through the bottom holes. It is this moving interior column of air that controls the flow of gasoline from the nozzle. Owing to the friction of the small passages, the speed of air flow through the sleeve does not increase as fast as the speed of air flow outside the sleeve, hence there is a tendency for the mixture to

remain constant. The throttle of this carburetor is of the barrel type, and the top of the spray nozzle and its surrounding sleeve are located inside the throttle.

In the more recent design shown at Fig. 106, provision is made for idling and altitude adjustment control. Air at atmospheric pressure enters the outside base or air tube of the diffuser column, passes up this outer

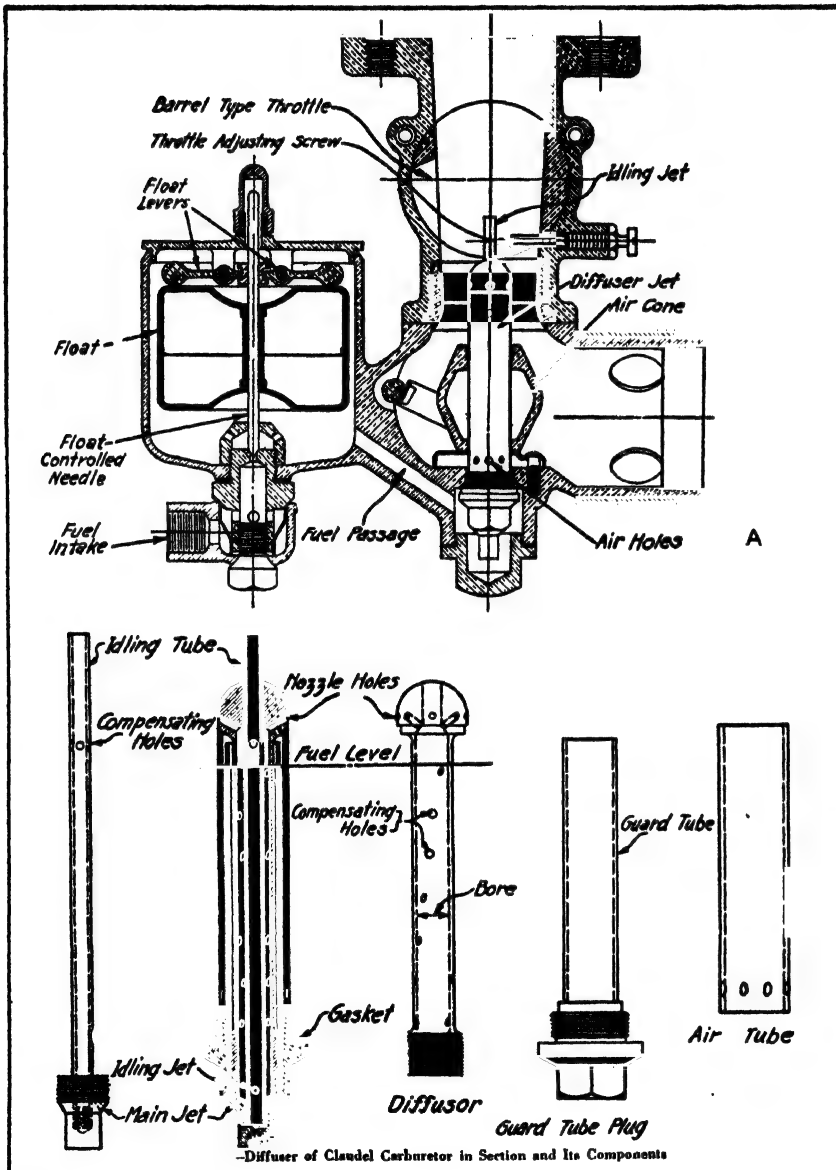


Fig. 106.—Sectional View of Claudel Aviation Type Carburetor Shows Arrangement of Internal Parts at A. The Diffuser of the Claudel Carburetor at B is Dismantled to Show its Components.

sleeve and over the top of the gasoline guard tube. As the suction in the diffuser increases, lowering the liquid level in the bore, a series of air bleed holes that provide compensation are progressively uncovered. The air rushes out into the ascending column of gas vapor and out of the nozzle holes at the top of the diffuser in a finely broken-up gasoline emulsion. The higher the suction acting upon the diffuser, the lower will be the level of gasoline within it, and therefore, more of the compensating holes will be uncovered, permitting a greater dilution of the mixture. At the higher speeds the diffuser is practically emptied and 21 spirally-arranged air

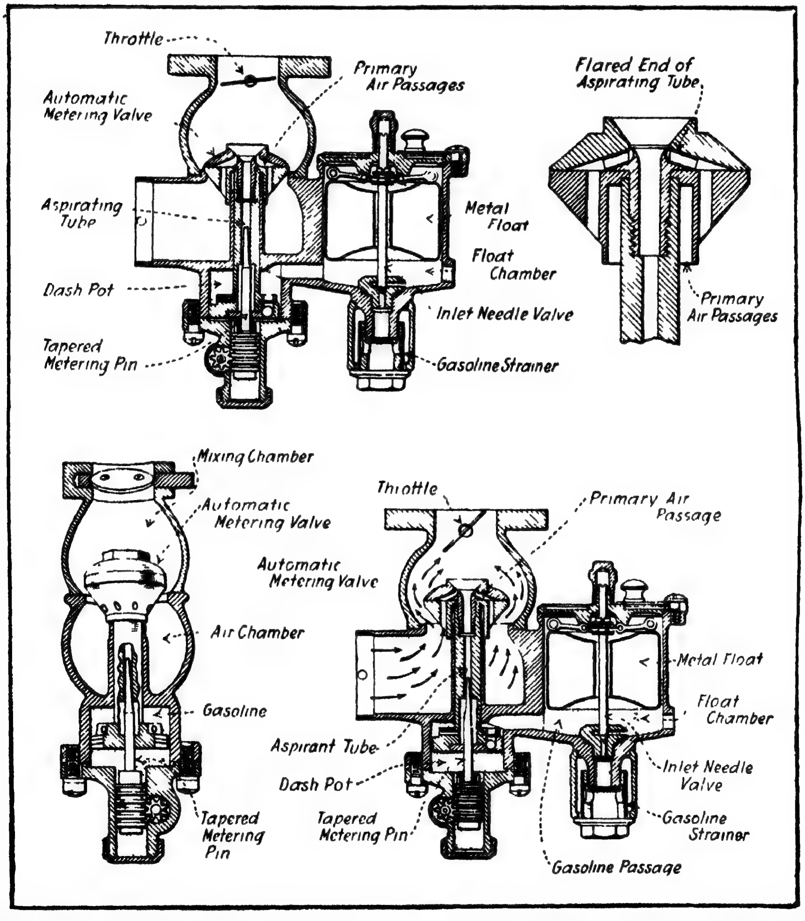


Fig. 107.—The Stewart Metering Pin Carburetor was Used on Some Early Aviation Engines.

bleed holes are in action. As the gasoline globules are lifted by the suction, they must pass through the air entering at right angles through the air bleed holes. This turbulent mixing effect produces a finely divided fuel emulsion. Any type of power or consumption curve desired can be secured by changing the size and position of the compensating holes in the diffuser wall.

The idling device is incorporated in a central tube projecting up into the depression of the barrel throttle where a strong pull is exerted on the idling jet for low speed action. The barrel throttle is slotted to pass around the idling jet and the only adjustment on the carburetor is a screw extending into the air space to partially block off the area of this slot as desired. Screwing it in lessens the air area and enriches the idling mixture. Screwing it out makes the mixture leaner.

On some models of the Claudel airplane carburetor the air cone is used for altitude compensation. In these models a venturi is installed several sizes in excess of the capacity required at ground level. The streamline cone is then raised and adjusted to meet the requirements at ground level and lowered to increase the air opening at altitudes as required.

Metering Pin Carburetor.—The carburetor shown at Fig. 107 is a metering type for automobile engines in which the vacuum at the jet is controlled by the weight of the metering valve surrounding the upright metering pin. The only moving part is the metering valve, which rises and falls with the changes in vacuum. The air chamber surrounds the metering valve, and there is a mixing chamber above. As the valve is drawn up the gasoline passage is enlarged on account of the predetermined taper on the metering pin, and the air passage also is increased proportionately, giving the correct mixture. A dash pot at the bottom of the valve checks flutter. In idling the valve rests on its seat, practically closing the air and giving the necessary idling mixture. A passage through the valve acts as an aspirating tube. When the valve is closed altogether the primary air passes through ducts in the valve itself, giving the proper amount for idling. The one adjustment consists in raising or lowering the tapered metering pin, increasing or decreasing the supply of gasoline. Dash control is supplied. This pulls down the metering pin, increasing the gasoline flow. The duplex type for eight- and twelve-cylinder motors is the same in principle as model 25, but it is a double carburetor synchronized as to throttle movements, adjustments, etc. The duplex for aeronautical motors is made of cast aluminum alloy.

Multiple Nozzle Vaporizers.—To secure properly proportioned mixtures some carburetor designers have evolved forms in which two or more nozzles are used in a common mixing chamber. The usual construction is to use two, one having a small opening and placed in a small air tube and used only for low speeds, the other being placed in a larger air tube and having a slightly augmented bore so that it is employed on intermediate speeds. At high speeds both jets would be used in series. Some multiple jet carburetors could be considered as a series of these instruments, each one being designed for certain conditions of engine action. They would vary from small size just sufficient to run the engine at low speed to others having sufficient capacity to furnish gas for the highest possible engine speed when used in conjunction with the smaller members which have been brought into service progressively as the engine speed has been augmented. The multiple nozzle carburetor differs from that in which a single spray tube is used only in the construction of the mixing chamber, as a common float bowl can be used to supply all spray pipes. It is common practice to bring the jets into action progressively

by some form of mechanical connection with the throttle or by automatic valves.

The object of any multiple nozzle carburetor is to secure greater flexibility and endeavor to supply mixtures of proper proportions at all speeds of the engine. It should be stated, however, that while devices of this nature lend themselves readily to practical application it is more diffcult to adjust them than the simpler forms having but one nozzle. When a number of jets are used the liability of clogging up the carburetor is increased, and if one or more of the nozzles is choked by a particle of dirt or water the resulting mixture trouble is difficult to detect. One of the

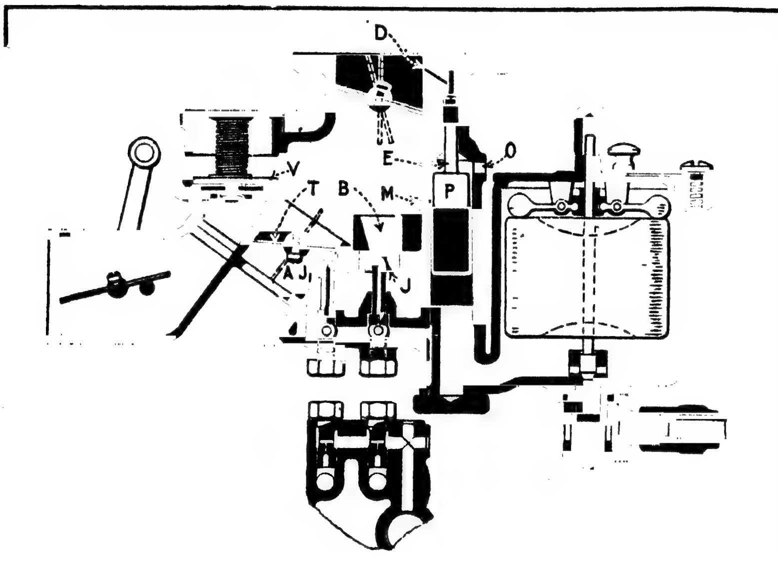


Fig. 107A.—The Ball and Ball Two Stage Carburetor, a Successful Automobile Type of Early Derivation.

nozzles may supply enough gasoline to permit the engine to run well at certain speeds and yet not be adequate to supply the proper amount of gas under other conditions In adjusting a multiple jet carburetor in which the jets are provided with gasoline regulating needles, it is customary to consider each nozzle as a distinct carburetor and to regulate it to secure the best motor action at that throttle position which corresponds to the conditions under which the jet is brought into service. For instance, that supplied the primary mixing chamber should be regulated with the throttle partly closed, while the auxiliary jet should be adjusted with the throttle fully opened.

Priming Necessary to Start Cold Engine.—The usual procedure in starting an engine cold is to supply a large excess of fuel (of the order of twenty times the normal proportion) so that the necessary amount of vapor can be obtained from the lighter fractions, the remaining part passing to the crankcase via the piston clearances, or directly to waste through the exhaust. Of the two evils, the latter is obviously preferable.

This excess is produced by priming, a *hand operated pump supplying fuel* directly to the induction manifold. Clearly, it is desirable that the method used for providing the excess fuel required on starting should not at the same time impair the metering characteristic over other parts of the range.

The presence of a stream of liquid fuel in the induction system is a complicating factor in the problem of carburetor operation. When the engine is running steadily at constant load and speed, the amount of fuel in the induction system remains constant, and the strength of the mixture supplied by the carburetor is the same as that received by the engine, though a number of cycles may occur during the passage of an individual drop of fuel in the liquid stream from the carburetor to the engine. When the load or speed is changed, the equilibrium is disturbed, and fuel has either to be added to, or taken away from, the manifold to restore equilibrium under the new conditions.

Function of Accelerating Wells.—During the transition period, the mixture strength supplied by the carburetor no longer corresponds to that received by the engine. It follows, therefore, that while the load or speed is changing, the whole of the induction manifold must be considered part of the carbureting system. To give good acceleration, the strength of the mixture received by the engine need never exceed that of the maximum-power mixture. The function of accelerating wells is to supply excess fuel to the manifold to enable the manifold to supply the correct mixture to the engine. Efficient accelerating wells assist economy of fuel, as they permit the normal use of mixtures weaker than would otherwise be required.

In the absence of suitable hot-spotting or other system of preheating, a considerable part of the vaporization of the fuel has to be carried out inside the engine cylinder. Under normal conditions, the greater part of the liquid fuel entering the cylinder is probably vaporized during the induction stroke, any residue being vaporized on the compression stroke. For this reason, in cold weather operation or in airplanes intended for high altitude work, some form of stove or heater worked in connection with the exhaust gas is considered desirable to heat the entering air, as will be described later.

Ball and Ball Two-Stage Carburetor.—This is a two-stage vaporizing device, hot air being used in the primary or initial stage of vaporization and cold air in the supplementary stage. Referring to the sectional illustration at Fig. 107 A it will be seen that there is a hot-air passage with a choke-valve; the primary venturi appears at B; J is its gasoline jet, and V is a spring-loaded idling valve in a fixed air opening. These parts constitute the primary system. In the secondary system A is a cold-air passage, T a butterfly valve and J a gasoline jet discharging into the cold-air passage. This system is brought into operation by opening the butterfly T. A connection between the butterfly T and the throttle, not shown, throws the butterfly wide open when the throttle is not quite wide open; at all other times the butterfly is held closed by a spring. The cylindrical chamber at the right of the mixing chamber has an extension E of reduced diameter connecting it with the intake manifold through a passage D. A restricted opening connects the float chamber with the cylindrical chamber

so that the gasoline level is the same in both. A loosely fitting plunger P in the cylindrical chamber has an upward extension into the small part of the chamber. O is a small air opening and M is a passage from the cylindrical chamber to the mixing chamber. Air constantly passes through this when the carburetor is in operation. The carburetor is really two in one. The primary carburetor is made up of a central jet in a venturi passage. The float chamber is eccentric. In the air passage there is a fixed opening, and additional air is taken in by the opening through suction of a spring-opposed air valve. The second stage, which comes

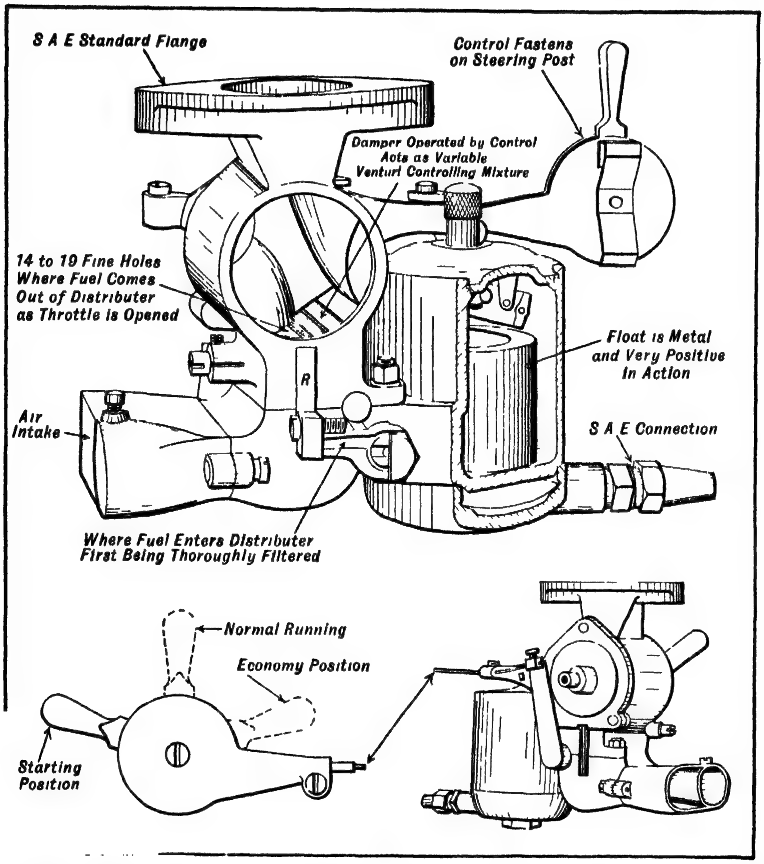


Fig. 108.—Sectional View Showing Construction of the Master Carburetor, a Multiple Nozzle Form Used on Racing Car Engines and Some Early Aviation Engines.

into play as soon as the carburetor is called upon for additional mixture above low medium speeds, is made up of an independent air passage containing another air valve. As the valve is opened this jet is uncovered, and air is led past it. For easy starting an extra passage leads from the float bowl passage to a point above the throttle. All the suction falls upon this passage when the throttle is closed. The passage contains a plunger and acts as a pick-up device. When the vacuum increases the

plunger rises and shuts off the flow of gasoline from the intake passage. As the throttle is opened the vacuum in the intake passage is broken, and the plunger falls, causing gasoline to gather above it. This is immediately drawn through the pick-up passage and gives the desired mixture for acceleration. Some early aircraft engines were fitted with carburetors of this type but it is not found on engines of recent development and is presented so the reader will understand the action of devices of this character.

Master Multiple-Jet Carburetor.—This carburetor, shown in detail in Figs. 108 and 109, has been very popular in racing cars and early aviation engines because of exceptionally good pick-up qualities and its thorough atomization of fuel. Its principle of operation is the breaking up of the fuel by a series of jets, which vary in number from fourteen to 21, according to the size of the carburetor. These are uncovered by opening

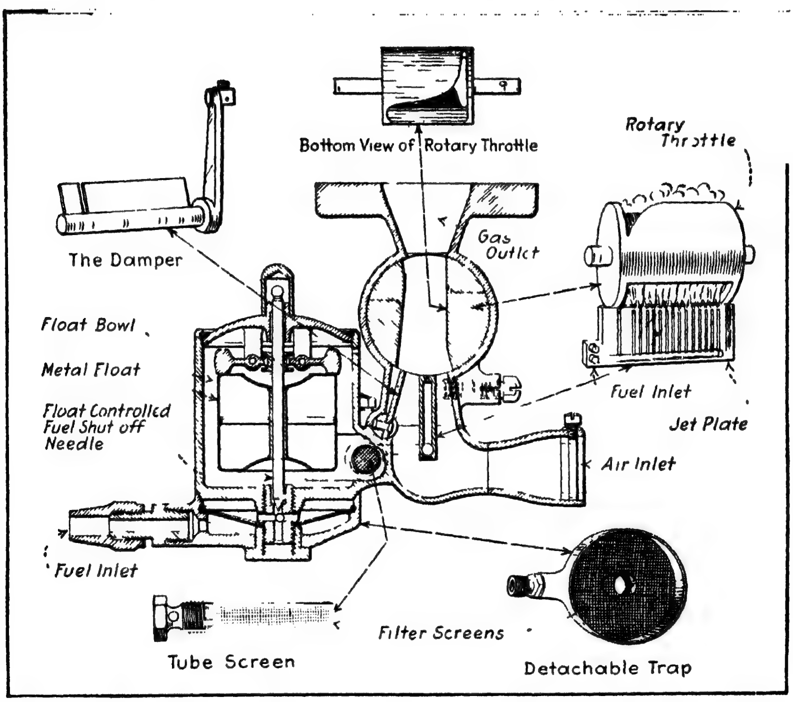


Fig. 109.—Outline Drawings of Various Master Carburetor Parts Showing the Location in the Main Assembly. Note Unusual Form of Spray Nozzles.

the throttle, which is curved—a patented feature—to secure the correct progression of jets. The carburetor has an eccentric float chamber, from which the gasoline is led to the jet piece from which the jets stand up in a row. The tops of these jets are closed until the throttle is opened far enough to pass them, which it does progressively. The air opening is at the bottom, and the throttle opening is such that a modified venturi is formed. The throttle is carried in a cylindrical barrel with the jets placed below it, and the passage from the barrel to the intake is arranged so that there is no interruption in the flow. For easy starting a dash-controlled

shutter closes off the air, throwing the suction on the jets, thus giving a rich mixture.

The only adjustment other than changing the jet plate and altering the float level is for idling, and once that is fixed it need never be touched. This is in the form of a screw and regulates the position of the throttle when at idling position. The dash control has high-speed, normal and rich-starting positions. In installing the Master carburetor the float chamber may be turned either toward the front or rear. If the float is turned toward the front a forward lug plate should be ordered; otherwise it will be difficult to install the control. The throttle lever must go all the way to the stop lug or maximum power will not be secured. In adjusting the idle screw it is turned in for rich and out for lean.

Notes on Adjustment of Simple Carburetors.—The modern float feed carburetor is a delicate and nicely balanced appliance that requires a certain amount of attention and care in order to obtain the best results. The adjustments can only be made by one possessing an intelligent knowledge of carburetor construction and must never be made unless the reason for changing the old adjustment is understood. Before altering the adjustment of any of the modern forms of carburetors, a few hints regarding the quality to be obtained in the mixture should be given some consideration, as if these are properly understood this knowledge will prove of great assistance in adjusting the vaporizer to give a good working proportion of fuel and air. There is some question regarding the best mixture proportions and it is estimated that gas will be explosive in which the proportions of fuel vapor and air will vary from one part of the former to a wide range included between four and eighteen parts of the latter. A one to four mixture is much too rich, while the one in eighteen is much too lean to provide positive ignition.

A rich mixture should be avoided because the excessive fuel used will deposit carbon and will soot the cylinder walls, combustion-chamber interior, piston top and valves and also tend to overheat the motor. A rich mixture will also seriously interfere with flexible control of the engine, as it will choke up on low throttle and run well on open throttle when the full amount of gas is needed. A rich mixture may be quickly discovered by black smoke issuing from the outlet stacks or exhaust ports, the exhaust gas having a very pungent odor. If the mixture contains a surplus of air there will be popping sounds in the carburetor, which is commonly termed "blowing back." To adjust a carburetor is not a difficult matter when the purpose of the various control members is understood. The first thing to do in adjusting a carburetor is to start the motor and to retard the sparking lever so the motor will run slowly leaving the throttle about half open. In order to ascertain if the mixture is too rich, cut down the gasoline flow gradually by screwing down the needle valve in those types where such jet orifice control is provided until the motor commences to run irregularly or misfire. Close the needle valve as far as possible without having the engine come to a stop, and after having found the minimum amount of fuel gradually unscrew the adjusting valve until you arrive at the point where the engine develops its highest speed. When this adjustment is secured the lock nut is screwed in place so the needle

valve will keep the adjustment and not vibrate loose.

The next point to look out for is regulation of the auxiliary air supply on those types of carburetors where an adjustable air valve is provided. This is done by advancing the spark lever and opening the throttle. The air valve is first opened or the spring tension reduced to a point where the engine misfires or pops back in the carburetor. When the point of maximum air supply the engine will run on is thus determined, the air valve spring may be tightened by screwing in on the regulating screw until the point is reached where an appreciable speeding up of the engine is noticed. If both fuel and air valves are set right, it will be possible to accelerate the engine speed uniformly without interfering with regularity of engine operation by moving the throttle lever or accelerator pedal from its closed to its wide open position, this being done with the spark lever advanced.

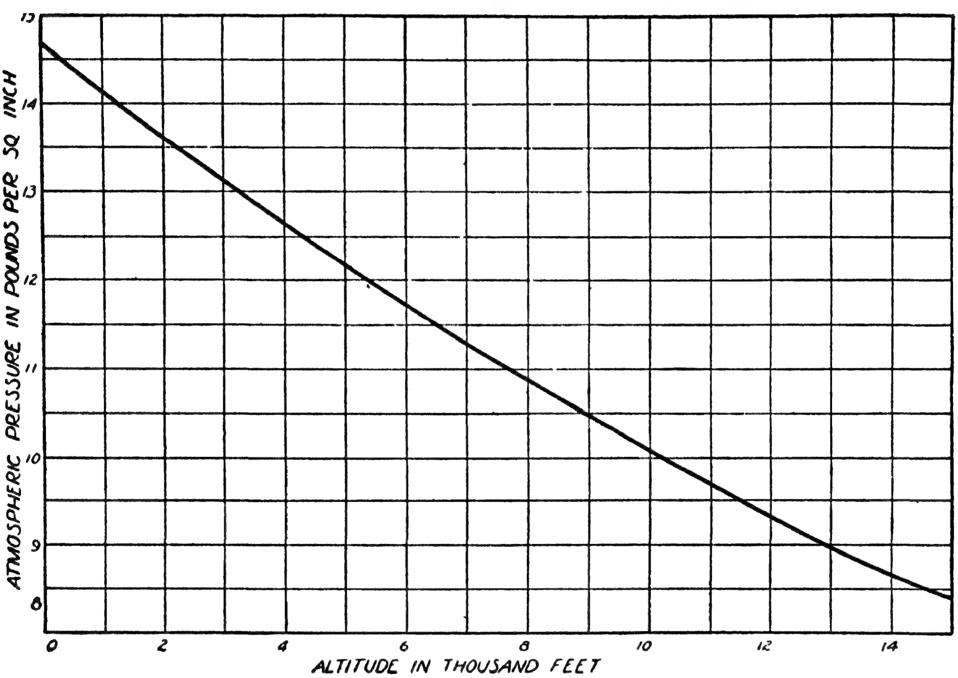
All types of carburetors do not have the same means of adjustment; in fact, some adjust only with the gasoline regulating needle; others must have a complete change of spray nozzles; while in others the mixture proportions may be varied only by adjustment of the quantity of entering air. Changing the float level is effective in some carburetors, but this should never be done unless it is certain that the level is not correct. Full instructions for locating carburetion troubles will be given in proper sequence. Most modern airplane engine carburetors can only be adjusted by supplying various combinations of choke tubes, venturi members and fixed bore jets. Needle valve regulation is found on automobile carburetors and in some few early aviation engines but is rarely used on modern engines. Metering plugs having various sizes of holes, as described in the chapter to follow are used to change proportions of fuel flow by using a plug with a smaller hole if too much fuel is supplied and vice-versa.

Effect of Altitude Changes.—It is a fact well known to experienced airmen that atmospheric conditions have much to do with carburetor action. It is often observed that a motor seems to develop more power at night than during the day, a circumstance which is attributed to the presence of more moisture in the cooler night air. Likewise, taking a motor from sea level to an altitude of 10,000 feet involves using rarefied air in the engine cylinders and atmospheric pressures ranging from 14.7 pounds at sea level to 10.1 pounds per square inch at the high altitude. All carburetors will require some adjustment in the course of any material change from one level to another. Great changes of altitude also have a marked effect on the cooling system of an airplane. Water boils at 212 degrees F. only at sea level. At an altitude of 10,000 feet it will boil at a temperature nineteen degrees lower, or 193 degrees F.

In high altitudes the reduced atmospheric pressure, for 5,000 feet or higher than sea level, results in not enough air reaching the mixture, so that either the auxiliary air opening has to be increased, or the gasoline in the mixture cut down. If the user is to be continually at high altitudes he should immediately purchase either a larger dome or a smaller strangling tube, mentioning the size carburetor that is at present in use and the type of motor that it is on, including details as to the bore and stroke. The smaller strangling tube makes an increased suction at the spray

nozzle; the air will have to be readjusted to meet it and one can use more auxiliary air, which is necessary. The effect on the motor without a smaller strangling tube is a perceptible sluggishness and failure to speed up to its normal crankshaft revolutions, as well as failure to give power. It means that about one-third of the regular speed is cut out. The reduced atmospheric pressure reduces the power of the explosion, in that there is not the same quantity of oxygen in the combustion-chamber as at sea level; to increase the amount taken in, you must also increase the gasoline speed, which is done by an increased suction through the smaller

VARIATION OF ATMOSPHERIC PRESSURE WITH ALTITUDE



(Society of Automotive Engineers)

strangling aperture. Some forms of carburetors are affected more than others by changes of altitude, which explains why the Zenith and Stromberg are so widely employed for airplane engine use. The compensating nozzle construction is not influenced as much by changes of altitude as the simpler nozzle types are and compensation means can be and are easily incorporated in the carburetor so the pilot can change the mixture proportions to allow for the reduced air density.

Compound Nozzle Zenith Carburetor.—The Zenith carburetor, shown at Fig. 110, has become very popular for airplane engine use because of its simplicity, as mixture compensation is secured by a compensating compound nozzle principle that works very well in practice. To illustrate this principle briefly, let us consider the elementary type of carburetor or mixing valve, as shown in Fig. 111 A. It consists of a single jet or spraying nozzle placed in the path of the incoming air and fed from the usual

float chamber. It is a natural inference to suppose that as the speed of the motor increases, both the flow of air and of gasoline will increase in the same proportion. Unhappily, such is not the case. There is a law of liquid bodies which states that the flow of gasoline from the jet increases under suction faster than the flow of air, giving a mixture which grows richer and richer—a mixture containing a much higher percentage of gasoline at high suction than at low. The tendency is shown by the accompanying curve (Fig. 111 B), which gives the ratio of gasoline to air

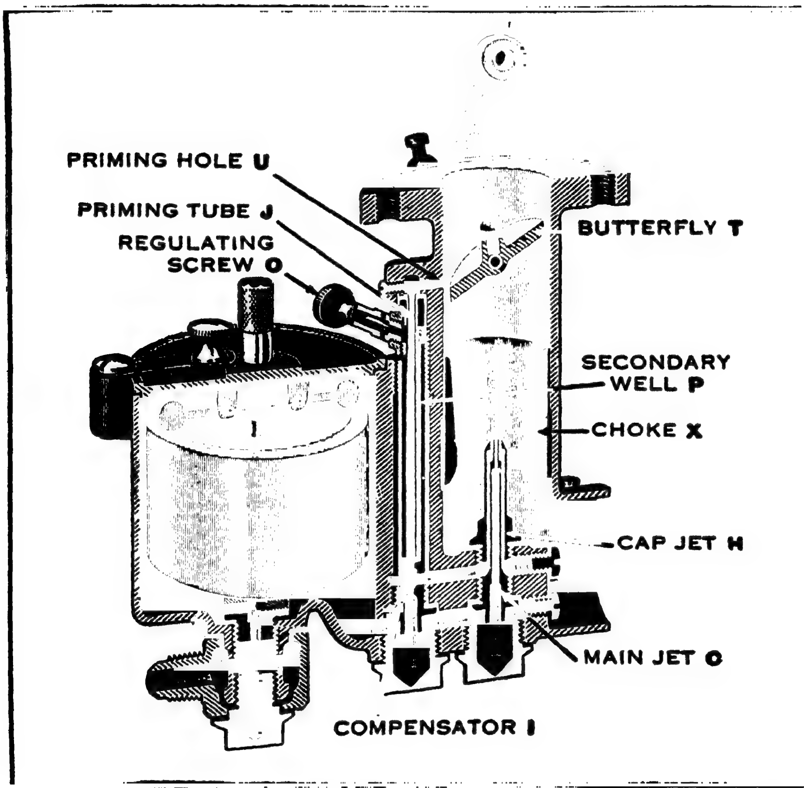


Fig. 110.—Sectional View of Zenith Compound Nozzle Compensating Carburetor of the Simple Type that was Widely used on Early Airplane and Automobile Engines.

at varying speeds from this type of jet. The mixture is practically constant only between narrow limits and at very high speed. The most common method of correcting this defect is by putting various auxiliary air valves which, adding air, tends to dilute this mixture as it gets too rich. It is difficult with makeshift devices to gauge this dilution accurately for every motor speed.

Function of Compensator.—Now, if we have a jet which grows richer as the suction increases, the opposite type of jet is one which would grow leaner under similar conditions. Baverey, the inventor of the Zenith, discovered the principle of the constant flow device which is shown in Fig.

111 C. Here a certain fixed amount of gasoline determined by the opening I is permitted to flow by gravity into the well J open to the air. The suction at jet H has no effect upon the gravity compensator I because the suction is destroyed by the open well J. The compensator, then, de-

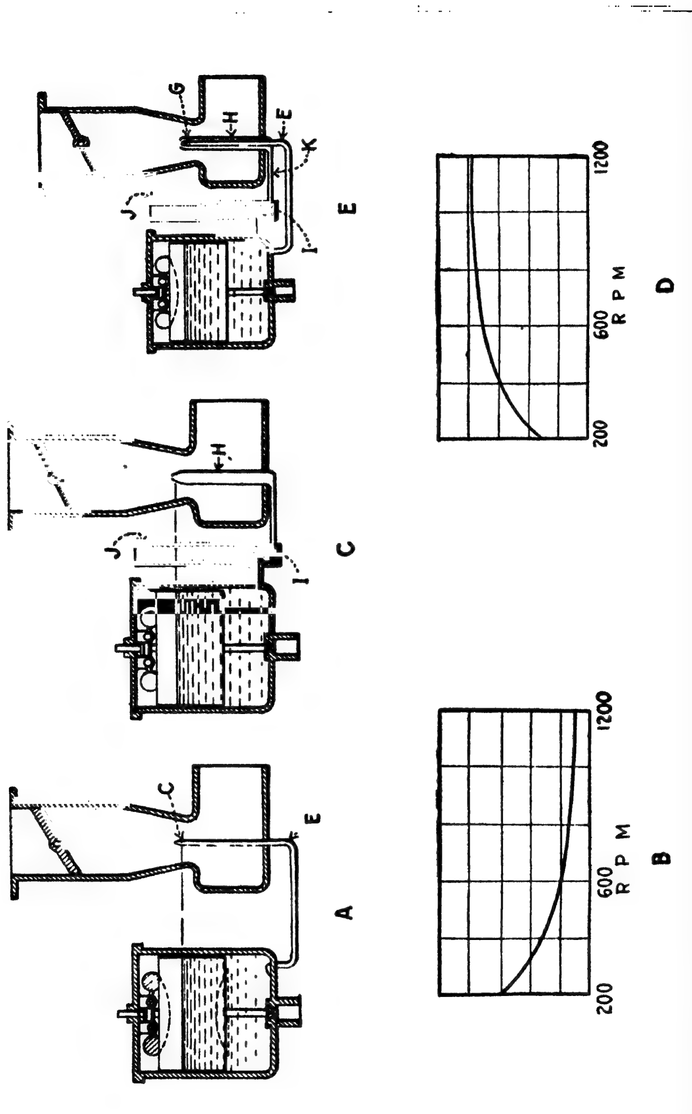


Fig. 111.—Diagrams Explaining Action of Bavary Compound Nozzle Used in Zenith Carburetors to Secure Automatic Mixture Compensation.

livers a steady rate of flow per unit of time, and as the motor suction increases more air is drawn up, while the amount of gasoline remains the same and the mixture grows poorer and poorer. Fig. 111 D shows this curve.

By combining these two types of rich and poor mixture carburetors the Zenith compound nozzle was evolved. In Fig. 111 E, we have both

the direct suction or richer type leading through pipe E and nozzle G and the "constant flow" device of Baverey shown at J, I, K and nozzle H. One counteracts the defects of the other, so that from the cranking of the motor to its highest speed there is a constant ratio of air and gasoline to supply efficient combustion.

In addition to the compound nozzle the Zenith is equipped with a starting and idling well, shown in the cut of Model L carburetor at P and J. This terminates in a priming hole at the edge of the butterfly valve, where the suction is greatest when this valve is slightly open. The gasoline is drawn up by the suction at the priming hole and, mixed with the air rushing by the butterfly, gives an ideal slow speed mixture. At higher speeds with the butterfly valve opened further the priming well ceases to operate and the compound nozzle drains the well and compensates correctly for any motor speed.

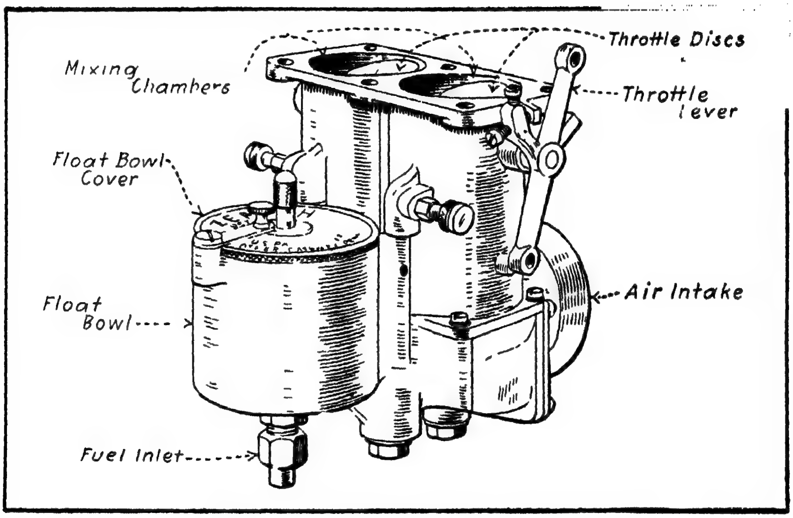


Fig. 112.—Zenith Duplex Carburetor for Airplane Motors of the "Vee" Type, Widely Used on War-Time Engines.

With the coming of the double motor containing eight or twelve cylinders arranged in two Vee blocks, the question of good carburetion has been a problem requiring much study. The single carburetor has given only indifferent results due to the strong cross suction in the inlet manifold from one set of cylinders to the other. This naturally led to the adoption of two carburetors in which each set of cylinders was independently fed by a separate carburetor. Results from this system were very good when the two carburetors were working exactly in unison, but as it was extremely difficult to accomplish this co-operation, especially where the adjustable type was employed, this system never gained in favor. The next logical step was the Zenith Duplex, shown at Fig. 112. This consists of two separate and distinct carburetors joined together so that a common gasoline float chamber and air inlet could be used by both. It does away with cross suction in the manifold because each set of cylinders has a

separate intake of its own. It does away with two carburetors and makes for simplicity. The practical application of the Zenith carburetor to the Curtiss 90 horsepower OX2 motor used on the JN4 standard training machine of wartime fame is shown at Fig. 112, which outlines a rear view of the engine in question. The carburetor is carried low to permit of fuel supply from a gravity tank carried back of the motor.

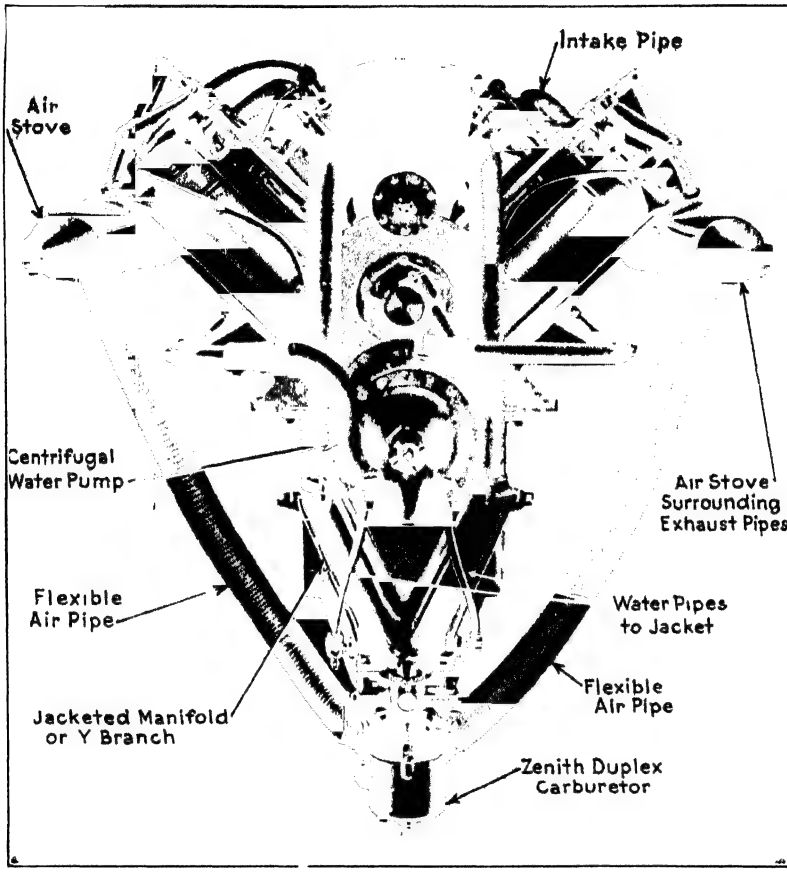


Fig. 113.—Rear View of Curtiss OX2, 90 Horsepower Airplane Motor Showing Zenith Carburetor Location, Design of Intake Manifold, which was Water Jacketed and Hot Air Leads from Air Stoves Surrounding Exhaust Pipes.

Zenith-Liberty Type.—The altitude adjustment of the Zenith Aeronautical Carburetor is illustrated diagrammatically at Fig. 114 A and as applied to the carburetor used on Liberty engines at Fig. 114 B. The float chamber is open to the air through two screened air inlets. The well J is in open communication at its top with the float chamber. A passage P is provided from the float chamber to the carbureting chamber below the throttle valve, this passage is fitted with a stopcock L which is manually operated from the pilot's seat. Under normal conditions, i.e., near the ground the stopcock should be closed. The fuel in the float chamber will be subjected to atmospheric pressure through screened air inlets.

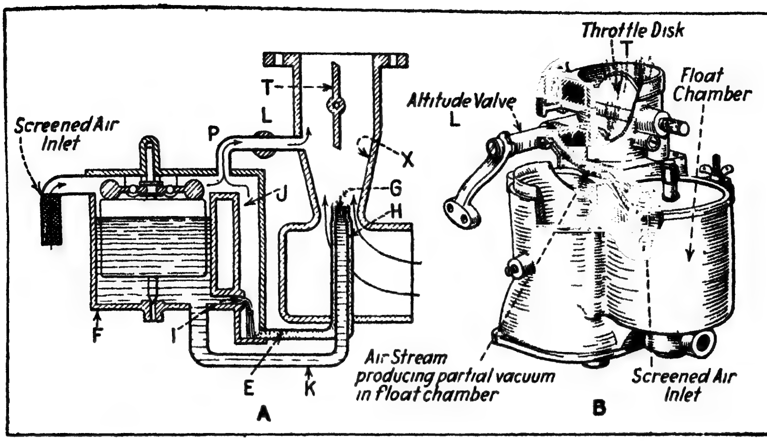


Fig. 114A.—Diagram Showing Principles Involved in Altitude Control of Zenith Aviation Carburetors. B—Practical Application of Altitude Control to Zenith Liberty Carburetor.

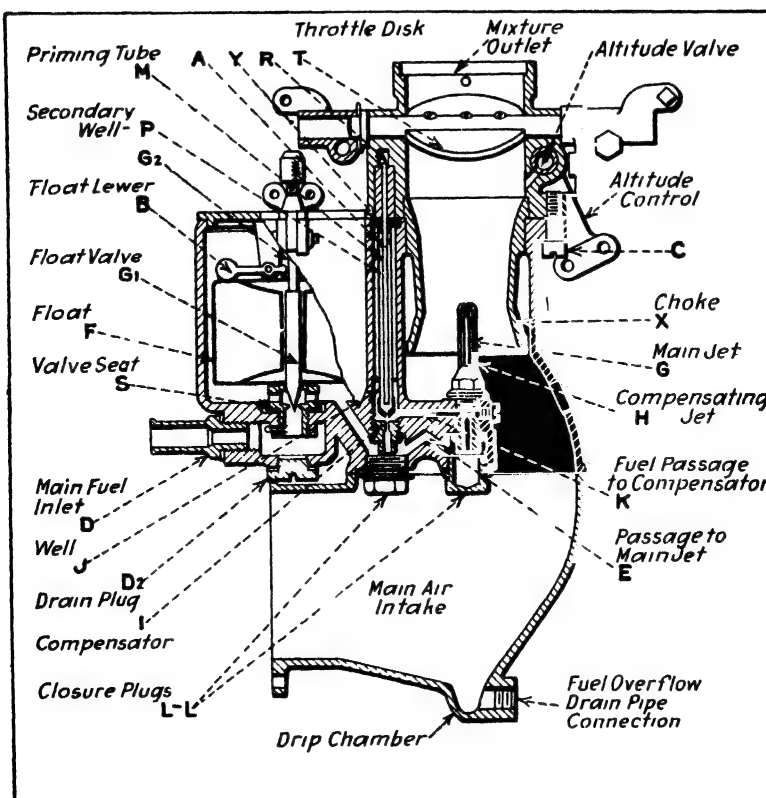
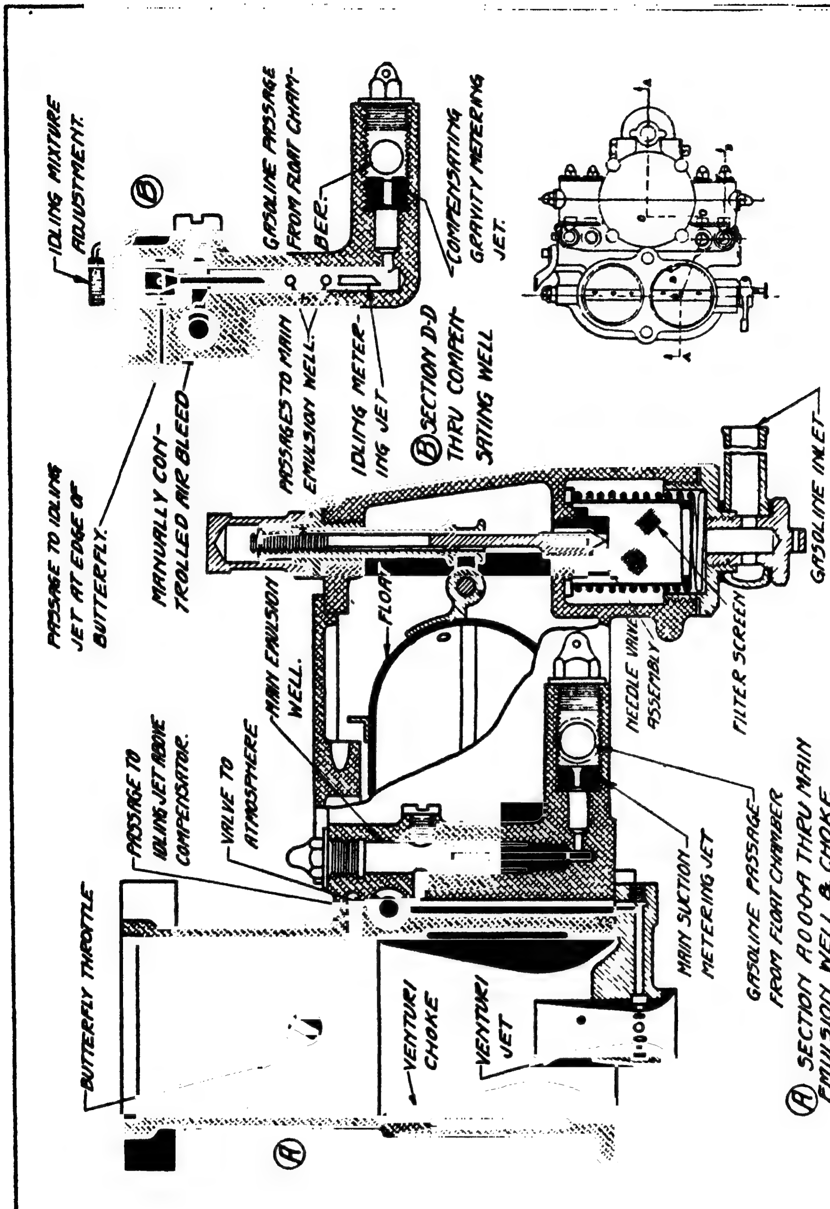


Fig. 114C.—Sectional View Showing Construction of Zenith-Duplex Carburetor Used with Liberty Aviation Engines.



Zenith Carburetor (Venturi Type Delivery Nozzle) Sectional View.

Fig. 115.—Zenith Aviation Type Carburetors with Venturi Type Delivery Nozzle.

When the engine is running, the partial vacuum produced in the choke X will draw fuel out of nozzles G and H in proper proportions. When an altitude of 6,000 feet is reached, the pilot will begin opening the valve L, thus drawing air from the float chamber and establishing a partial vacuum therein, this depending upon the amount the valve L is opened. The partial vacuum or suction effect on top of the gasoline in the float chamber will reduce the flow through the jets because of its retarding effect and.

the mixture will become more lean. The altitude valve should be opened as much as possible without producing a drop in the engine r.p.m.

The carburetor shown at 114 C is the unit used on Liberty engines, two being used, each one feeding six cylinders, though as they are duplex types, a separate mixing chamber is provided for each three cylinders. The proportions of the mixture are carefully determined by three variables, the choke, jet and compensator and should not be altered after brake tests have shown that the sizes selected are best for the motor to which the carburetor is fitted.

The numbers stamped on each of these parts indicate the size. The chokes are numbered according to the smallest inside diameter in millimeters. The jets and compensators are numbered according to the diameter of the opening in hundredths of a millimeter. For instance, a No. 105 jet has a hole 1.05 millimeters in diameter. The choke and jet sizes which have been found to give the best results on Liberty "12" engines are:

Choke	No. 31
Jet	No. 145
Compensator	No. 155
Idling Jet	No. 70

Zenith with Venturi Type Delivery Nozzle.—Another type of Zenith carburetor having a Venturi type delivery nozzle is shown at Fig. 115. From the usual type of float chamber the gasoline passes through two metering jets into two wells. The first, or main emulsion well is under the depression of the venturi nozzle which draws the fuel through the jet. The second, or compensating well is maintained at atmospheric pressure by an air bleed, consequently the gasoline flow through the compensating jet is by gravity only. The two wells communicate by means of two passages. The delivery to the venturi nozzle is from the main emulsion well only. In operation, the flow through the main metering jet increases more rapidly than the engine speed and if this jet alone was depended on the mixture would become richer with increasing speed. The compensating jet is made too lean to run the engine alone and the flow is constant at all speeds. The flow through the compensating jet decreases about as much as that in the main jet increases as motor r.p.m. augment and as both jets work together the proper compensation results. For slow running the vertical idling tube receives air and gasoline from the compensating well. The mixture richness is controlled by raising or lowering the conical screw shown at B.

The "Bristol" Triplex Carburetor.—This carburetor has been specially designed for the "Bristol" Jupiter engine and consists of three carburetors combined in one unit, operated by one set of controls, and necessitating only one fuel feed pipe in place of the three pipes used in a conventional design. This design, besides simplifying construction, gives a saving of weight of approximately 50 per cent, and results in a very compact instrument, which can be mounted low enough to obtain a good gravity feed and yet be placed inside the cowling. Each carburetor has its own independent float chamber, jets, mixing chamber, choke and throttle. To suit the

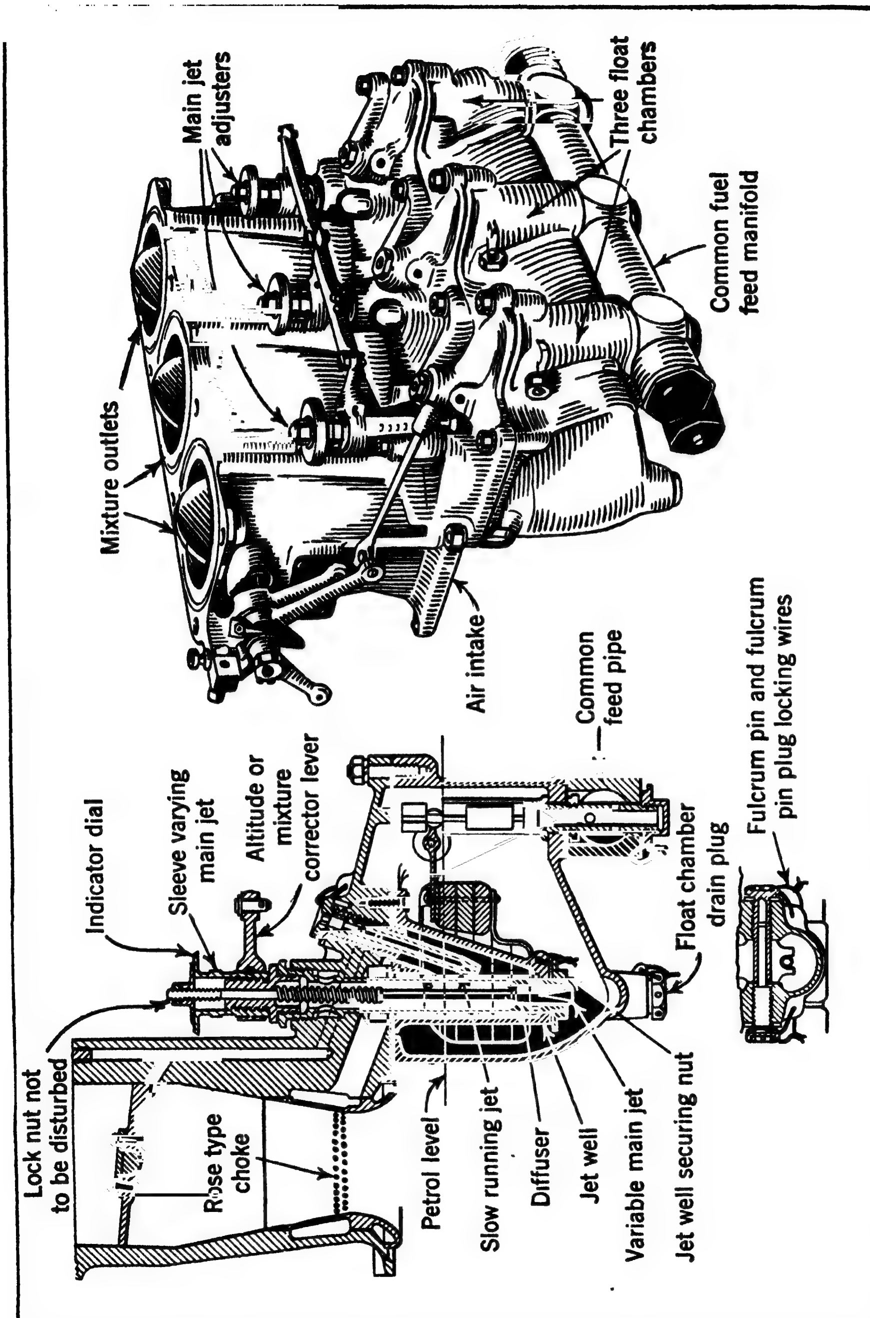


Fig. 115A.—The Bristol Triplex Carburetor Used on Bristol-Jupiter Engines Has an Adjustment for the Main Jet.

convenience of installation the throttle and altitude control levers are adjustable to any angular position, and are inter-connected in order to ensure the altitude lever being automatically returned to the rich or ground position when closing the throttle on a glide or dive, thus preventing the possibility of the engine being damaged through running at ground-level on the weakened setting used for altitude work.

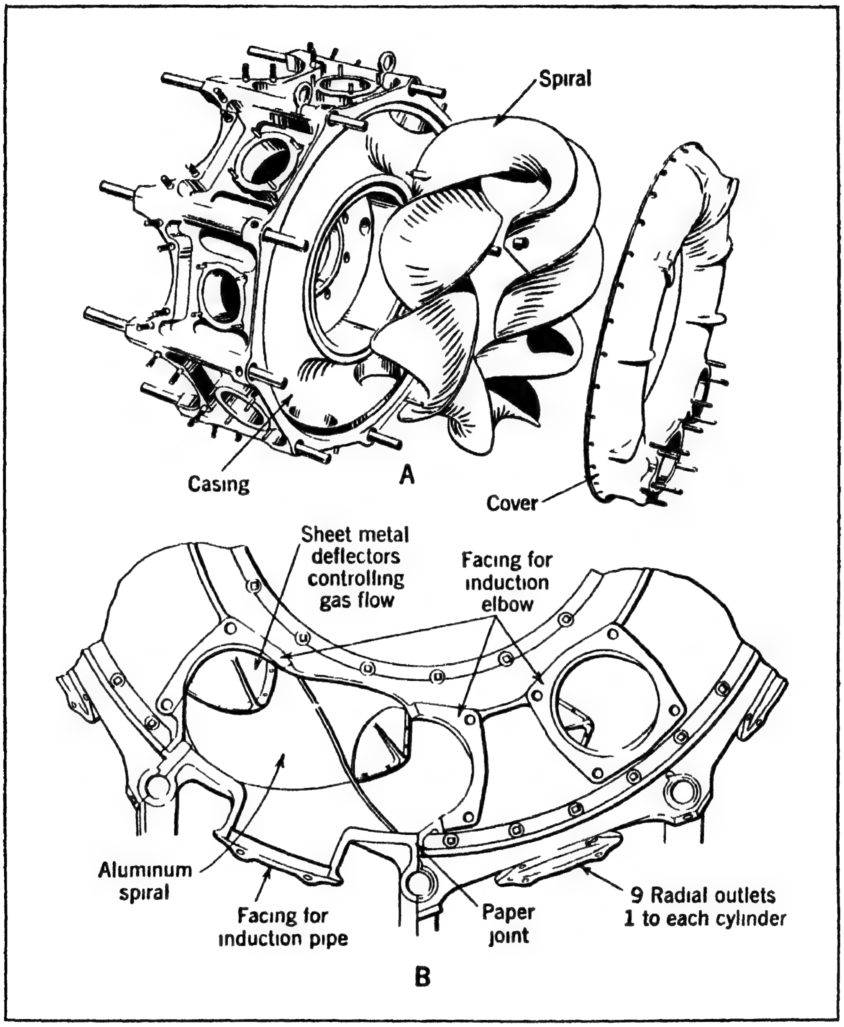


Fig. 115B.—The Bristol Jupiter Series VI Induction Spiral.

The main jet is of the variable type, greatly facilitating tuning, and permitting its use as a simple form of altitude control, giving an exceptional range, as borne out by the results of official tests on high altitude scouts. For commercial purposes, the main jet can be controlled in flight to give the maximum fuel economy under cruising conditions, resulting in a 25 per cent saving in fuel consumption. The choke and air balance have been

designed to give stable operating characteristics over a wide range of conditions, permitting the bench setting being utilized without returning for varying installation flight conditions. The float mechanism is simple but efficient and reliable, and is arranged to pass sufficient fuel to cope with the maximum demand with an ample margin, and to give freedom from flooding under the greatest possible variation in normal operating altitudes. As can be seen by consulting the sectional view at Fig. 115 A, the fuel from the main jet passes into the entering air stream through a series of holes in the restricted throat of the venturi. The main jet adjustment is by simple mechanism, a lever actuated internally threaded sleeve lifting a control stem against spring pressure. The float is made up of a number of cork pieces suitably fastened together. The fuel control needle, actuated by the float is a simple counterbalanced form with liberal area of seating. In view of the explanatory matter which has been given on carburetor action, it will not be difficult to understand the functioning of the Jupiter triplex fuel mixture producing device.

The Bristol Jupiter Series VI Induction Spiral.—A patented feature that was incorporated in the Bristol Jupiter models included in Series VI made for more uniform mixture distribution in nine-cylinder radial engines. This spiral distributor, which is shown in Fig. 115 B, ensured a homogeneous mixture and contributed materially to the smooth running of the engines built prior to the incorporation of the gear driven blower included in the latest models. The spiral is a circular three start casting housed in the annular induction chamber formed in the rear of the crankcase, the chamber being closed by a spiral cover. The three starts of the spiral form, with the spiral chamber, three separate channels, each isolated from the other and fed by one carburetor, each channel feeding three evenly spaced cylinders. This arrangement ensures that even in the event of any interference with the functioning of one carburetor, the engine will continue to run smoothly. The deflectors ensure that the gases rotate in the desired direction, and are arranged so that even in the event of a back fire the distribution is not upset.

QUESTIONS FOR REVIEW

1. What is the simplest practical form of carburetor and how does it work?
2. Why was the float feed carburetor developed?
3. What is a metering valve carburetor?
4. What advantage does a multiple jet carburetor have over a single jet form?
5. Why are accelerating wells needed?
6. What is the effect of altitude change on mixture adjustment and how is it compensated for?
7. Outline action of Zenith Compound nozzle.
8. Describe altitude adjustment on Zenith-Liberty type.
9. How does the Bristol triplex carburetor work?
10. What is the function of the induction spiral?

CHAPTER X

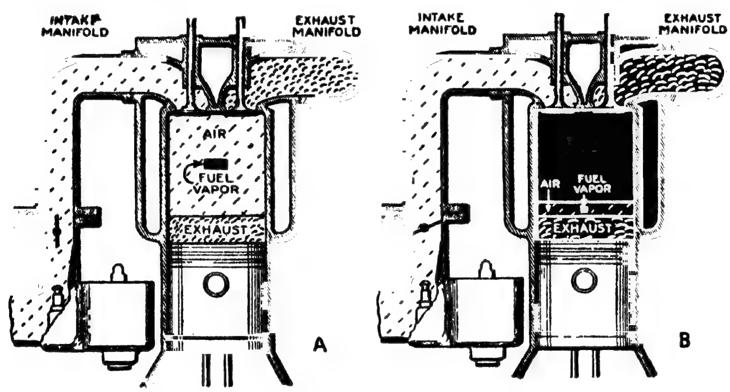
STROMBERG AVIATION CARBURETORS

The Air Charge—Requirements of Firing Mixture—Effect of Valve Overlap—Suction Pulsations and Blowback—The Venturi Tube—Effects of Pulsations Upon Average Suction—Stromberg Aircraft Carburetors—Carburetors Differ on Various Engines—The Plain Jet and the Air Bleed—The Idling System—The Accelerating System—Actual Arrangement of Parts—Fuel Supply and Float Action—Operation in Different Airplane Positions—Function of the Strainer—Float Valve Construction—Interchangeability of Float Parts—The Fuel Jet Systems—The Main Jet System—The Main Discharge Assembly—Volume of Accelerating Well—Idling Jet Adjustment—Idling Adjustment—Altitude Mixture Control—Float Chamber Suction Control—Airport Control—Combined Airport and Float Suction Control—The "S" Series—The Double Models.

The Air Charge.—The air flow through the carburetor is a result of the downward motion of the pistons of the engine on their suction stroke, which induces a suction and corresponding air flow through the carburetor and intake manifold into the engine cylinder. The carburetor and manifold air passages are made large enough to admit freely the amount of air required to fill the cylinder at full speed or, in other words, large enough to fill the cylinders with air at atmospheric pressure and nearly atmospheric density by the time the end of the suction stroke is reached. The power of the engine is controlled and reduction of speed obtained by use of a throttle valve, which regulates the admission of air to the engine according to the extent of its opening. When the throttle valve is partly closed, less air can flow past it to fill the space vacated by the downward stroke of the pistons, and there is consequently a strong suction or partial vacuum in the intake manifold and cylinder. On the other hand, as the throttle valve is closed, there is less suction below it in the carburetor, since less air is being drawn through. At full speed and full open throttle the suction or partial vacuum in the intake manifold above the carburetor is only about .4 to .8 pound, while at the minimum speed, idling, the partial vacuum is between 7 and 9½ pounds per square inch; the values given are for sea level, where the external atmospheric pressure is about 14½ pounds per square inch.

Requirements of Firing Mixture.—It is generally assumed that any quantity of air in the cylinder will ignite provided it has mingled with it gasoline or fuel vapor of $\frac{1}{16}$ to $\frac{1}{11}$ of its own weight. This is not strictly true, however, under the conditions which exist when the engine is idling. On the exhaust stroke preceding the suction stroke of the engine, the piston does not occupy the compression or clearance space above the limit of its travel and this space is consequently left filled with exhaust gas at nearly atmospheric pressure at the time the intake valve opens and the suction stroke begins. The presence of this unburnable exhaust gas has little effect upon the rapidity or certainty of combustion when a full air charge is admitted to the cylinder but when the throttle is nearly closed and only a small amount of fresh air is admitted on each suction stroke, the ignition becomes very slow or may not take place at all. If, for instance, the two

blocks of cylinders of a Vee-type engine are fitted with different carburetors and if the throttle of one block is open to a position corresponding to 300 revolutions per minute while the other throttle is open to a 700 r.p.m. position, so that the engine is actually turning 500 r.p.m., the amount of air reaching the cylinders of the first mentioned block will be too thin for normal combustion, and the ignition or burning of the charge may be so slow that it is still continuing when the exhaust valve opens, showing a flame of red fire from the exhaust pipes; or a number of the cylinders may not fire at all. If several ignitions fail, more and more unburned mixture will accumulate in the clearance space until after several revolutions there will be enough air for one explosion and the engine will, therefore, fire intermittently, with strongly marked torque vibration.



Relative amounts (shown in volumes at atmospheric pressure) of air, gasoline vapor and exhaust gas in cylinder at beginning of compression. FULL POWER, FULL OPEN THROTTLE.

Relative amounts of air, gasoline vapor, and exhaust gas in cylinder at beginning of compression. ENGINE IDLING, THROTTLE CLOSED. Actually the cylinder contents expand to fill the whole space, at less than half atmospheric pressure.

Fig. 116.—Diagram Showing Relative Amounts of Air and Fuel Vapor for Full Power and Engine Idling Throttle Position.

Trouble of this sort is very common with engines having more than one carburetor unit, and can only be cured by accurate and positive adjustment or "synchronization" of the several throttle valves. It should be noted that air leaks in the intake manifold joints or exhaust valves that do not seat tightly will unbalance multiple carburetor systems in the same way, even though the gasoline feed for idle be adjusted as accurately as possible. Fig. 116 A and B show the normal proportions of air, fuel and exhaust gas in the mixture charge, while Fig. 117 A and B illustrate the effect of air and exhaust gas leaks.

Effect of Valve Overlap.—A condition equivalent to that caused by air leaks in the intake manifold is obtained when there is a perceptible valve overlap or duration of time at the top of the piston travel between the exhaust and intake stroke when both intake and exhaust valves are open.

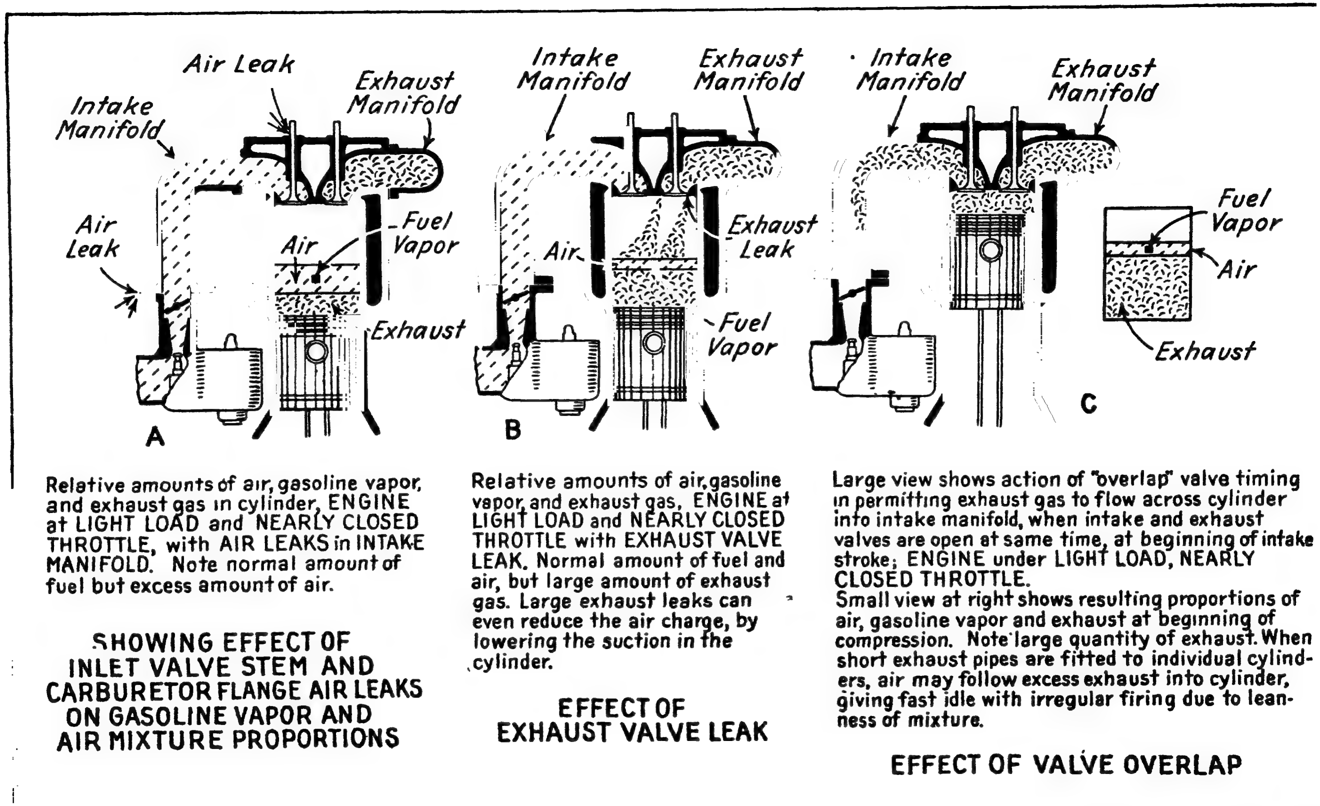


Fig. 117.—Diagrams Showing Relative Amounts of Air, Gasoline, Vapor, and Exhaust Gas Under Various Engine Operating Conditions. A—Nearly Closed Throttle with Air Leaks in Intake Manifold. B—Nearly Closed Throttle with Exhaust Leak. C—Action of Overlapping Valve Timing.

In this case, as shown in Fig. 117 C, there can be an actual flow of exhaust gas and air into the combustion-chamber and intake manifold, which air feed will make leaner the mixture in the cylinder whose suction stroke comes next. When the throttle is open, the amount of air entering the manifold by this route is small compared with the amount of air coming through the carburetor, but when the throttle is nearly closed, the percentage of air coming in through the valve overlap may be considerable. The idling adjustment of these carburetors has sufficient range to give enough extra fuel to take care of this increased air flow. If, however, the valve overlap is different on different cylinders—usually due to the valve tappet or rocker arm clearance not being uniform—the cylinders may all

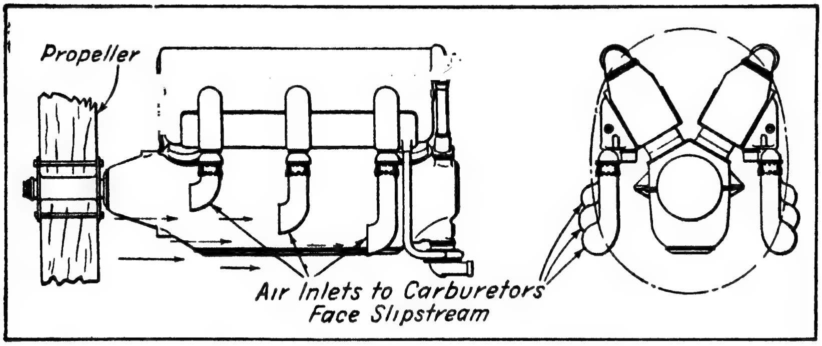


Fig. 117D.—Method of Installing Air Inlet Scoops when Multiple Carburetors are Used, in Order to Prevent Obstructing Air Flow.

get different strengths of air charge, which, in conjunction with the unchanged fuel flow from the carburetor, will result in each cylinder having a different strength of idling mixture. Engines whose valves are operated by push rods often have a variable amount of overlap, depending on the temperature of the cylinders if the expansion of the cylinder is different from that of the rod; this being perhaps most marked with aluminum cylinders and air-cooled engines. Under such conditions, the valve overlap will be different with the engine cold from that with it warm and the difference in air taken in through the valve overlap may not only affect the mixture proportion but also the speed of idling. If a large amount of air is taken in from this source with the engine cold, the engine will idle faster, and on a leaner mixture than for the same throttle closing with the engine hot; and, for satisfactory results, it will be necessary to set the idling mixture adjustment to take care of the cold engine condition, even though this results in a very rich low speed mixture when the engine is warm. It is obvious that when there is considerable valve overlap, smooth low speed operation can only be obtained by having the valve tappet or rocker arm clearances set exactly at the proper value and uniform on the different cylinders.

Suction Pulsations and "Blowback."—A suction stroke lasts only a little longer than one-half of a crankshaft rotation and in any given cylinder occurs every second turn or fourth half turn. With fewer than four cylinders drawing from one carburetor opening the suction strokes are, there-

fore, separated by intervals during which there is no actual demand for air in the engine. The resulting sudden interruption of high velocity air flow through the intake manifold often develops a temporary rebound of air or "blowback" through the carburetor. This is the cause of the cloud of fuel spray often noticed at the air entrance of carburetor units feeding three cylinders. Waste of fuel from this source can be avoided by making the air intake to the carburetor of sufficient length. Turning the air scoop toward the propeller blast as shown at Fig. 117 D helps but does not entirely obviate this feature.

As will be noted in later paragraphs, suction pulsations, in conjunction with the use of the throttle valve, tend to disturb the natural metering characteristics of the carburetor.

The Venturi Tube.—It is a fortunate and useful result of natural laws that both the air flow through an opening of fixed size and the fuel flow through an "air bleed jet" system respond in substantially equal proportion to changes of suction (within the range of air velocities used in the carburetor). To maintain an approximately uniform mixture proportion throughout the power range of engine operation, it is only necessary that the "air bleed jet" and metering air opening be exposed to the engine suction in the same degree, which condition is obtained by locating the fuel jet outlet in the center of a definitely formed air nozzle or venturi tube (sometimes called the "choke"), both being on the atmospheric side of the throttle valve. The venturi tube has another use than this, however. As previously stated, full power output from the engine requires that the manifold suction or partial vacuum be low, between .4 and .8 pounds, at full engine speed, when the suction below the throttle valve is the maximum. From the standpoint of metering and spraying the fuel, it would be desirable to use a suction several times this. It has been found that both these requirements can be complied with by the use of the peculiarly shaped air passage of a venturi tube, consisting of a reduced or constricted central portion with a smooth round entrance and a gradually tapered outlet. With this it is possible to obtain, on a jet located in the central portion, several times the suction existing in the intake passage beyond the venturi tube, and thereby maintain a low manifold vacuum with a high fuel metering suction.

As the venturi tube constitutes the limitation of air capacity of the carburetor, it is made in different sizes which may be selected according to the requirements of the engine to which the carburetor is fitted. The size is usually selected such that at normal full speed and load, there will be a mean air velocity (during the suction stroke) of 300 feet per second through the throat or narrowest part. This air velocity should correspond to a mean partial vacuum at the mouth of the carburetor, with throttle full open, of about sixteen inches water during the suction stroke; that is, with the carburetor supplying four or more cylinders the vacuum just above the carburetor would read on a water gauge sixteen inches, for three cylinders twelve inches and for one cylinder four inches.

Effect of Pulsations upon Average Suction.—The pressure drop, or suction, in a moving column of air is, within the limits of carburetor operation, proportional to the square of the air velocity. When the velocity

varies, the suction at each moment will be proportional to the square of the velocity at that moment, but the average of this suction, as registered by a water gauge or by the flow from a jet, will be greater than the square of the *average* velocity, as a trial computation with assumed values will show. For this reason, strongly marked pulsations of air flow, such as exist when fewer than four cylinders draw from one carburetor, will show a higher mean suction in the venturi, and draw more fuel from the jet, than would be the case were the same amount of air drawn through the carburetor at a uniform rate. The effect of the pulsations on the carburetor action is strongest at wide open throttle, for as the throttle is closed, it has a tendency to dampen or partly smooth out these pulsations at the discharge nozzle. Therefore, a carburetor which would give a uniform

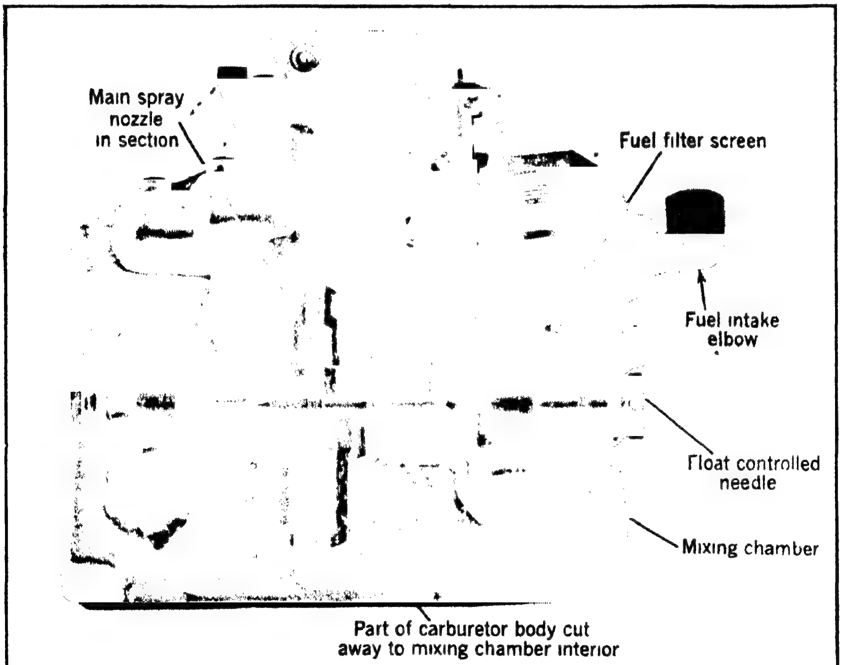


Fig. 118.—Part Sectional "Cutaway" View of Stromberg NA-T4 Aviation Engine Carburetor Showing Fuel Supply Nozzle and Filter Screen Surrounding Float Control Fuel Supply Opening.

mixture over a range of throttle positions with smooth air flow, would thin out as the throttle was closed under pulsating flow, due to the decreasing effect of the pulsations causing less fuel to flow in proportion to the air. On certain engines this tendency has been so marked as to require a special jet form for its correction.

Stromberg Aircraft Carburetors.—The new series of Stromberg airplane carburetors described in the following pages have been developed by recent and exhaustive tests in the laboratory and in flight service and their construction represents an improvement over types previously manufactured. They give smooth engine operation throughout the range from idling speed

to full throttle. An idling adjustment is provided to give a minimum idling speed. The throttle can be opened rapidly from any position with the engine warm or cold and an immediate response is obtained without misfires or backfires. The mixture delivered at full throttle is that required for maximum power; at intermediate throttle positions it is that of maximum economy. The carburetor action is unaffected by the changes in airplane position encountered in ordinary flying and the carburetors will continue to supply the engine with fuel when the airplane is "stunted." The mixture control provided on all models is smooth in its action and yet is powerful enough to give correction for high altitudes. Some models embody an automatic mixture control which relieves the pilot of all necessity of making adjustments. In design, the carburetors are compact and sturdy with all necessary adjustment parts easily accessible. The material

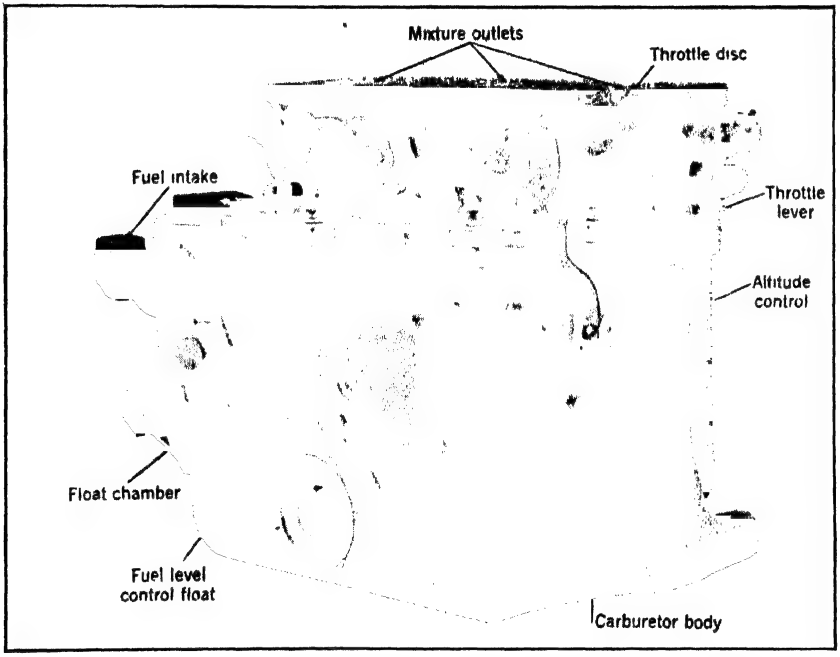


Fig. 119.—Part Sectional View of Stromberg NA-T4 Aviation Engine Carburetor Showing Float and Altitude Adjustment Control.

and workmanship are held to a high standard. The carburetors are thoroughly tested throughout the process of manufacture and before leaving the factory are given a final flow test, a duplication of operation on an engine.

Carburetors Differ on Various Engines.—In the present stage of aircraft development, the requirements of engine design are often so rigorous as to demand a special carburetor for each model of engine. The Stromberg Motor Devices Company maintains a standard line of single vertical models which is applicable to many types of engines. Each twin, or duplex type, has usually been designed to fit one particular engine. Owing to the number of different models, it has been impossible to give a detailed descrip-

tion of each one in this book, but, as these models are all substantially similar in principle and construction, the information furnished will be found generally applicable. Detailed instructions as to proper jet sizes, fuel level settings, etc., will be found in the instruction books of the engines upon which each of these carburetors is used, and further information will be furnished gladly by the makers upon request.

All Stromberg aircraft carburetors carry the general model designation "NA." Following a hyphen (-), the next letter indicates the type, "S" for single vertical carburetor, "U" for double vertical carburetor with float chamber between the barrels, "Y" for double carburetor with a double float chamber fore and aft of the barrels, "L" for inverted type, and so on. The final numeral indicates the nominal rated size of the carburetor, the sizes starting from one inch which is number one, and increasing in ¼-inch steps. For example a two-inch carburetor is number five. The actual diameter of the carburetor barrel opening is $^3/_{16}$ inch greater than the nominal rated size, in accordance with the standards of the Society of Automotive Engineers. A final letter is often used to designate a special series of one particular model. The model designation and serial number will always be found marked on the carburetor. On the earlier carburetors this infomation is carried on the aluminum name plate which is riveted to the main body, usually on the top of the float chamber. On the later carburetor the model designation and serial number are stamped on a cast boss on the main body of the carburetor.

For example, the carburetor shown at Figs. 118 and 119 is the Stromberg NA-T-4 and is a three outlet type used on Wright Whirlwind engines, J5A and J5B models. We will now proceed to a consideration of the basic principles of Stromberg airplane carburetors.

The Plain Jet and the Air Bleed.—It is generally believed that a simple plain fuel jet in a carburetor air opening of fixed size tends to deliver a continuously richer mixture as the engine suction and air flow increase, but this is not accurately true. Under the suctions of medium and high engine speeds, as carburetors are now built, a plain jet will give a fairly uniform mixture; but coming down to low speeds and suctions, the jet delivery falls off very markedly in relation to the air flow. This is due to the fact that some of the suction force is consumed in raising the fuel from the float level to the jet outlet (to avoid overflow with motor not running, the jet must necessarily stand a safe distance above the fuel level), and in overcoming the tendency of the fuel to adhere to the jet tip. At low suctions, the discharge from a plain jet is as shown in Fig. 120 A, with the fuel clinging to the metal of the jet and tearing off intermittently in large drops. The discharge from a plain fuel jet is, therefore, retarded by an almost constant force, which is insignificant at high suctions, but which perceptibly reduces the flow at low suctions. The application of the "Air Bleed" principle in overcoming this difficulty is shown in the accompanying illustrations. Fig. 120 B shows a familiar instance of how suction may be great enough to lift a liquid above its level, without drawing any of it away. Now if a tiny air hole be pricked in the side of the straw above the liquid surface and the same suction applied as before, bubbles of air will enter the straw and the liquid will be drawn up in a continuous series of

small slugs or drops, as shown in Fig. 120 C.

Such a construction is not quite suitable for a carburetor jet, as there is still a distance through which the liquid must be lifted from its level, before the air begins to pick it up; also the free opening of the straw at its bottom prevents very great suction being exerted on the air bleed hole or vent, just as too large an air opening in proportion to the straw size would reduce the suction available to lift the liquid. A modification to take care of these points is shown in Fig. 120 D, in which the air is taken in

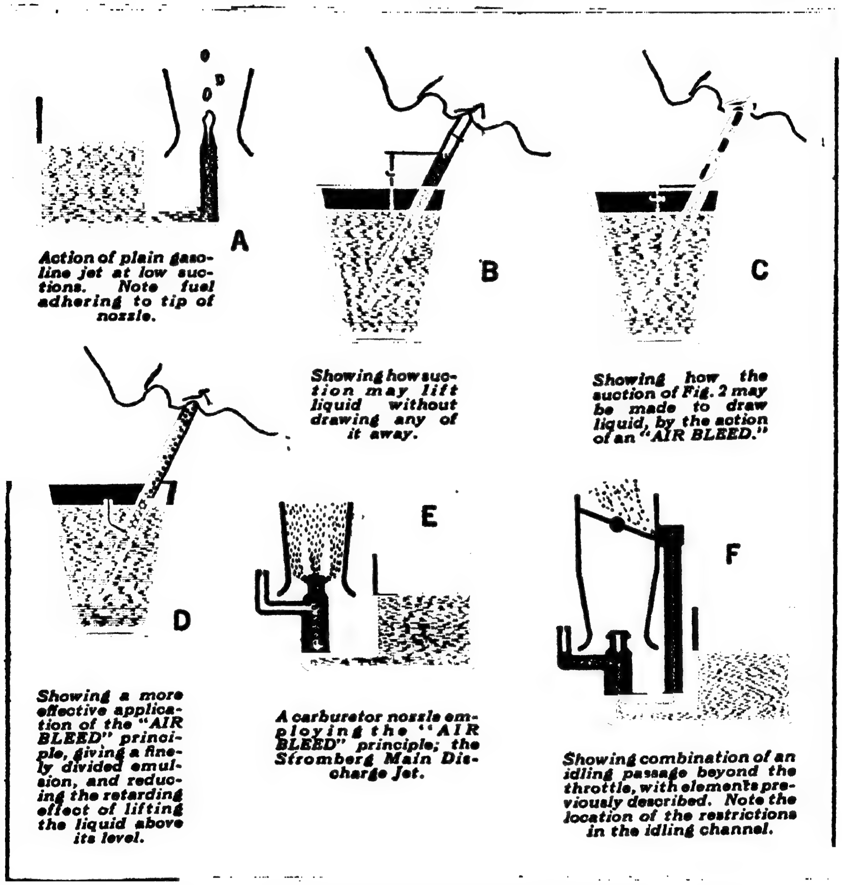


Fig. 120.—Diagrams Explaining Application of Air Bleed Principle to the Stromberg Main Discharge Jet.

slightly below the liquid level and a restricting orifice placed at the bottom, with the result that a finely divided emulsion of air and liquid is formed in the tube. The construction just described, when incorporated into a carburetor jet, takes the form shown in Fig. 120 E. Such a jet tends to give a substantially uniform mixture under steady speed throughout its range of operation. The mixture proportion can also be modified for high speed and low speed as desired by proper selection of dimensions of air bleed and emulsion channels.

The Idling System.—The structure of Fig. 120 E does not entirely meet the requirements of carburetor service because at low engine speeds the air flow does not have sufficient force to carry the fuel up from the jet to the throttle valve. As shown in Fig. 120 F a bypass or idling passage is provided to carry the fuel up to the throttle valve and intake manifold when the main jet suction is weak. Above idling speeds the fuel metering is controlled primarily by the main jet suction, the idling system being mainly a transfer or bypass independent of the main jet metering system. When the main jet suction is low, the fuel discharges through the idling system and as the main jet suction increases, some fuel will begin to deliver therefrom. The actual application of these principles to the carburetor is shown at Fig. 121, which is a sectional diagram showing general construction of Stromberg aircraft carburetors.

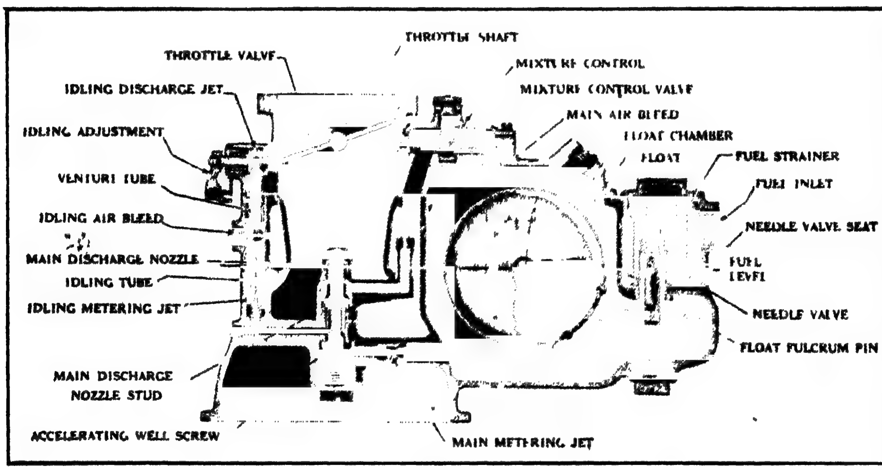
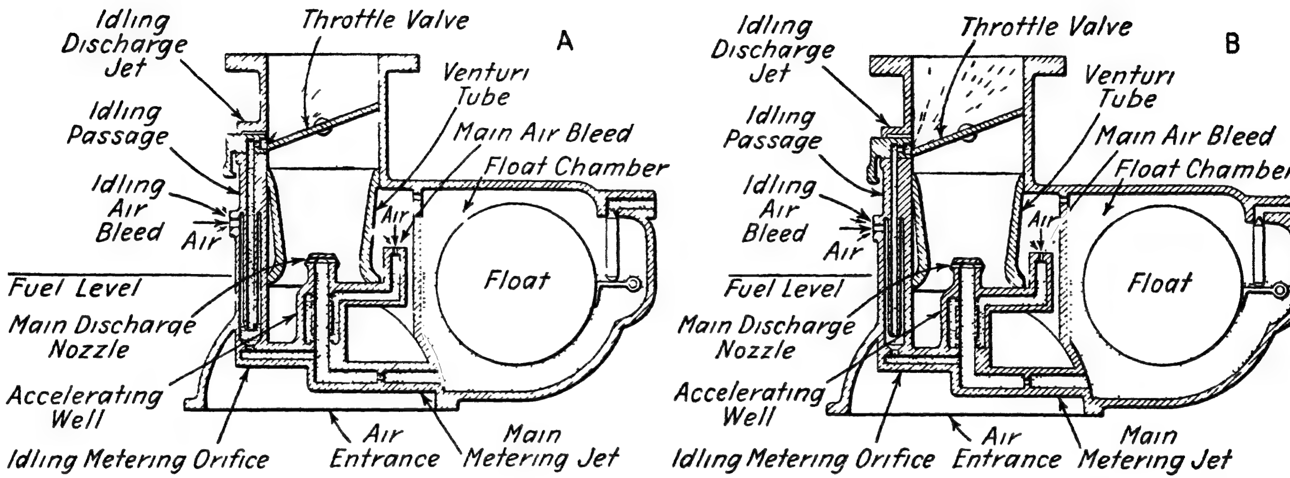


Fig. 121.—Lettered Diagram Showing General Construction of Stromberg Aircraft Carburetors with All Important Parts Indicated.

The Accelerating System.—It will be obvious that quick changes of engine speed and throttle position would involve rapid reversals of fuel flow through this idling system, tending toward temporary periods of lean mixture. It has been found that these may be avoided by the use of an "Accelerating Well" which is merely a downward extension or enlargement of the air bleed passage. The depression or suction in the central channel is always greater than in the outer or "Well" chamber, and any increase in suction on the main jet results in a lowering of the level in the well chamber. The volume of fuel thus displaced temporarily supplements the fuel delivered through the metering orifice, covering up any lag in either the idling tube or main jet passages, and gives a rich mixture when the throttle is opened quickly from low speeds. Such a rich charge is of especial value in obtaining a prompt response from the engine when the throttle is opened after a long glide during which the engine has become cold. The accelerating well system has no effect upon the mixture proportion except during change of speed or load.



Starting Note that idling passage draws unrestricted gasoline flow (without "AIR BLEED") from space around idling tube, thus giving temporarily rich mixture and automatic priming action, when this space is emptied and engine starts, the idling "AIR BLEED" begins to operate as in B

Engine idling Fuel flow of normal mixture strength taken through restriction of idling metering orifice, joined by air of idling "AIR BLEED" and fed in fan-shaped spray across mouth of carburetor above throttle Note accelerating well nearly full.

Fig. 122.—Diagram Showing Fuel Flow in Stromberg Carburetor when Starting at A and with Engine Idling at B.

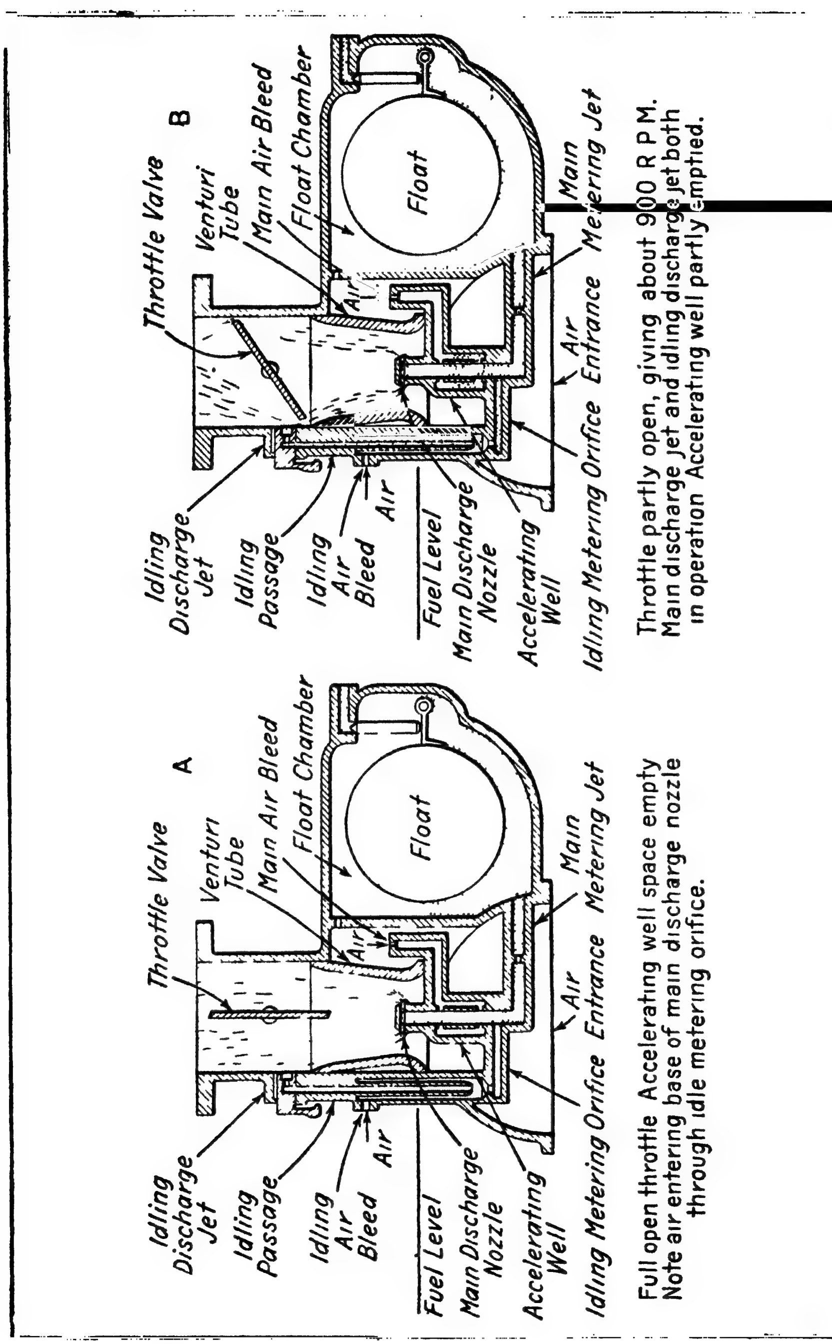


Fig. 123.—Diagram Showing Fuel Flow in Stromberg Carburetor at Full Open Throttle at A, and at Partially Open Throttle at B.

Actual Arrangement of Parts.—Figs. 122 and 123 show in diagrammatic form and greater detail the arrangement actually employed in the carburetor. The fuel level is maintained by a float and shut-off valve. The float vent and main jet air bleed are located in a recess of the air entrance of the carburetor so the metering suction on the fuel jet will not be disturbed or modified by the propeller blast or other dynamic air pressure around the engine. For reasons of structural convenience, the accelerating well space is made concentric around the main discharge jet fuel passage. The venturi tube is made as a removable bushing so that its size may be selected according to the air capacity of the engine. The fuel metering jet is also made removable so that various sizes may be selected as necessary to give the fuel flow desired. The main jet air bleed size exerts very little effect upon the mixture and seldom need be changed. In order that the delivery through the idling system may increase somewhat with the throttle opening, the idling discharge opening is located at the edge of the throttle valve, so that opening of the throttle will increase the area subject to manifold depression and thus give greater flow through the idling system. As generally fitted, the idling metering jet meters the fuel from 250 r.p.m. to 700 r.p.m., above which speed the mixture is governed by the main metering jet size.

The idling system also contains an air bleed which serves the threefold purpose of reducing the suction on the idling metering orifice to controllable limits, of providing a convenient means of mixture regulation, and of contributing to the operation of the priming device. The idling passages are made of considerably larger size than would be necessary if they were carrying fuel only, the suction through them being reduced in normal running by the idling air bleed. While the engine is at rest, the fuel rises to the float level both inside and outside the idling tube, this combined space being made equal to the volume of a rich fuel charge for one cylinder. In starting as at Fig. 122 A if the throttle be closed, the first quarter turn of the propeller will draw this rich charge into the intake manifold before the air bleed flow through the idling jet system can begin. If an interval of a few seconds be allowed for refilling the idling tube, another quarter turn will draw in another rich charge, and so on. Thus the carburetor automatically primes the engine for starting. If the engine is so warm that this priming action is not desired, opening the throttle one-fourth of its way will reduce the manifold vacuum so much that no priming action will take place as the propeller is turned over.

When the throttle is nearly closed, in a position corresponding to the lowest idling speed as shown at Fig. 122 B the idling mixture may be controlled in two ways: first, by changing the location of the idling discharge jet so that it shows more or less opening above the throttle; second, by changing the size of the idling air bleed. When the throttle is opened so that its edge has passed the idling discharge jet as at Fig. 123 B the idling mixture can then be controlled only by the idling air bleed. The earlier Stromberg models were made with the idling adjustment in the form of a taper needle controlling the size of the idling air bleed and this adjustment affected not only the idling speed, but sometimes the mixture as high as 1,200 r.p.m. of the engine. Most of the later models have the idling air

bleed set as a fixed size hole in a removable plug, the idle adjustment being obtained by change of the idle discharge jet to show more or less of its *opening* above the throttle edge; the effect of this adjustment rarely carries above 600 r.p.m.

Fuel Supply and Float Action.—In the airplane carburetor the fuel flow should be subject to no other force than the suction generated by the air flow through the carburetor. It therefore is necessary that there be in the carburetor, between the main gasoline tank and the fuel jets, a separate constant level reservoir or float chamber. The action of the float mechanism is indicated in Fig. 124 A, which shows the design used in many Stromberg carburetors. With no fuel in the carburetor the float drops

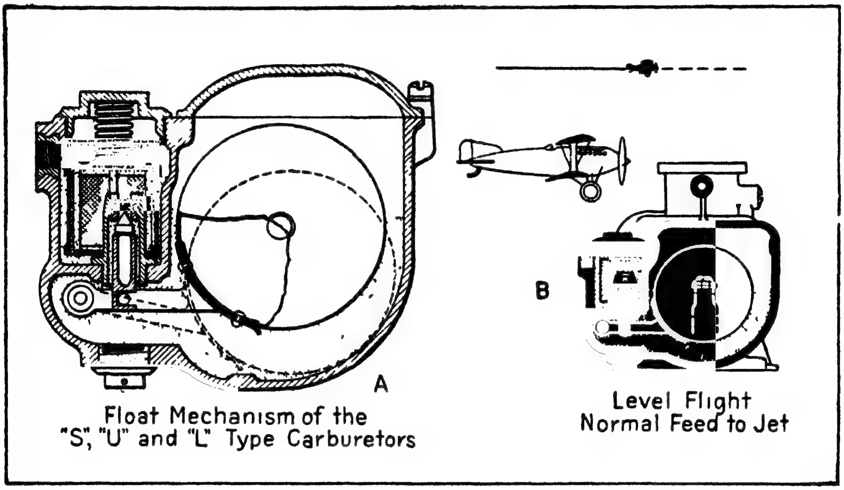


Fig. 124.—Float Mechanism of Stromberg Aircraft Carburetors at A. Fuel Feed in Level Flight at B.

down to the position shown by the dotted lines, leaving the needle valve open. As fuel is admitted from the supply line—passing through the strainer before reaching the float chamber—the float rises and shuts the valve as the fuel reaches the level shown. When the motor is running and fuel is being drawn out of the float chamber to the jets, the valve does not alternately open and close, but takes an intermediate position such that the valve opening is just sufficient to keep the fuel supplied and the level constant. This running level is usually about one-eighth inch below the standing level. While the engine is operating, the vibration usually keeps the fuel splashing considerably above the running level. These float mechanisms have been designed to operate at between one and four pounds pressure, though they will generally feed sufficient fuel at eighteen-inch gravity head and withstand six pounds pressure without flooding with engine not running; they will usually stand nine to ten pounds pressure without flooding sufficiently to affect the operation of the engine when running. The NA-ZD5 model, with an especially large float designed for the Packard "Shenandoah" engines, will stand greater pressures than those given above.

Operation in Different Airplane Positions.—In airplane service it is

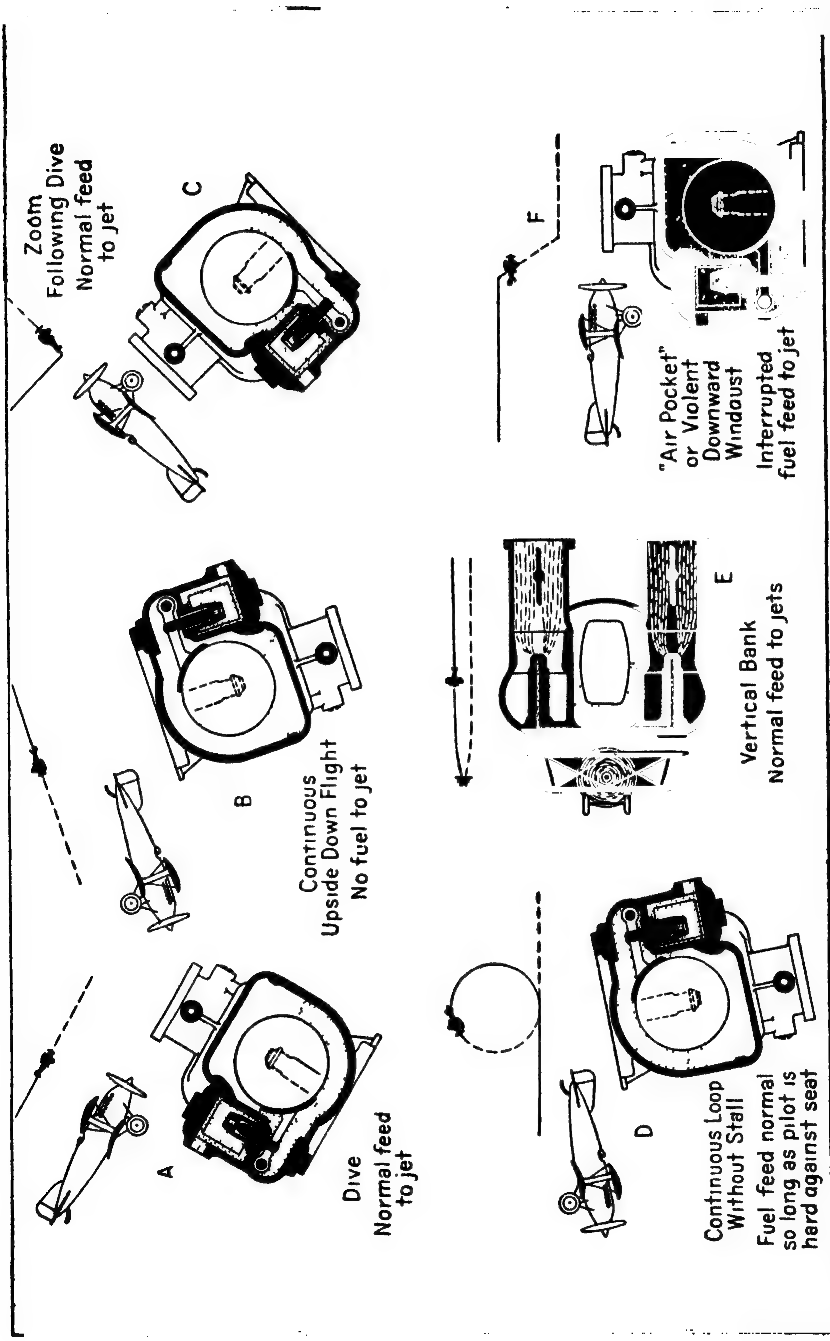


Fig. 125.—Fuel Level in Stromberg Aircraft Carburetor for Various Airplane Positions. A—Dive. B—Continuous Up-Side-Down Flight. C—Zoom, Following a Dive. D—Continuous Loop without Stall. E—Vertical Bank. F—Air Pocket or Violent Downward Wind Gust.

necessary that this mechanism should operate positively at all angles and positions where power is demanded from the engine, also that it should not permit leakage of gasoline in other positions. The three views, Fig. 124 B and Figs. 125 A and B, show the conditions in these carburetors in several different positions. During a dive, climb or side-skid, the action is normal, due to the way the float is suspended. When the engine is stalled with plane upside down, the float is no longer supported by the gasoline, and the valve shuts off as shown in Fig. 125 B. The operation of the float mechanism and the position of the fuel during different aerial maneuvers depend not only upon gravity, but also upon the motion and position of the airplane. The motion of the airplane involves inertia and, during certain movements, centrifugal force, while the position of the airplane determines the position of the outlets from the float chamber relative to the earth. Figs. 125 C, D, E and F show the position of the fuel in the float chamber during certain maneuvers, assuming that the carburetor is mounted with the float pivot toward the rear of the airplane. The ultimate result can be expressed simply in terms of the pilot's sensation of position, since his body is acted on by the same forces as the fuel mass. The carburetor float will function normally whenever the pilot is resting on his seat, leaning hard against the back or the sides of his seat, or tending to slide forward. If the position or motion of the plane is such that the pilot tends to leave his seat and be supported by the life belt, the same forces will cause the float to go up, that is, to close the float needle valve. At the same time the fuel will go to the top of the float chamber and cease to flow from the discharge nozzles. This action can occur when a violent gust of wind forces the airplane down so quickly that the pilot leaves his seat. In such case, the fuel will take the position shown in Fig. 125 F and temporarily cease to flow from the discharge nozzles, even though the airplane be right side up.

In older airplane carburetor practice, the carburetor barrels and fuel discharge nozzles were located ahead of or behind the float chamber; with such an arrangement, when standing with tail down, or diving at a steep angle, the main jet was considerably above or below the fuel level. When above, there was a tendency for the mixture to be unduly lean; when below, there was a tendency for the fuel to leak out. In the newer Stromberg types, the fuel discharge nozzles are located in line laterally with the center of the float, with the result that the fuel flow is not disturbed in any normal flying position. In the NA-Y5 Duplex type, which was designed for use in the limited lateral space of the Curtiss D-12 engine, two floats are used, both attached to the same lever and valve, one ahead of and one to the rear of the fuel discharge jet. As the carburetor is inclined, the fuel level rises on one float and goes down on the other but its position with reference to the discharge nozzle is not changed.

Function of the Strainer.—In all models, the fuel supply first enters a strainer chamber where it must pass through the strainer screen, which intercepts any dirt particles which might clog the needle valve opening or, later, the jets. The strainer is retained by the strainer plug and a compression spring, and can be readily removed when the strainer plug is taken out. A drain plug is also provided on the left side of the strainer chamber (in designating the left and right sides of a carburetor, it is always assumed

that the carburetor is held with the gasoline connection toward the speaker). By removing this plug the strainer chamber can be thoroughly drained and flushed out.

Float Valve Construction.—The float valve parts on an airplane engine are subject to a severely destructive effect from the engine vibration, and, therefore, receive the most careful construction and selection of materials. In these carburetors the needle valve seat and float lever pin are of hard Monel metal, while the needle point is made of an even harder noncorroding alloy steel. This graduation in hardness is used in order that the unavoidable wear may be confined to the seat, so that it will tend to conform to the shape of the needle. The level reached by the fuel in the float chamber depends somewhat upon the specific gravity, being slightly higher as the fuel is lighter. As set at the factory these float valves will operate properly and hold the level sufficiently close, with fuels ranging between 58 to 76 degrees Baumé gravity. Alterations in the level are obtained by the use of thicker or thinner gaskets under the needle valve seats. Proper selection of gaskets and convenient methods of measuring and setting the level and of removing and replacing the different parts, are described later.

Interchangeability of Float Parts.—The same float needle valve, needle seat, strainer and strainer plug and fulcrum screw have been used in nearly all models described. The NA-ZD5 carburetor, whose particular service required a smaller fuel opening in the needle valve seat, has this a No. 12 drill size. The standard size of this opening is No. 9 drill size. The NA-Y5 model has a special needle and seat with a No. 20 drill size opening. Due to limitations of space, it has been necessary to use different shaped floats in different models. The NA-D4, NA-L5 and NA-S8 carburetors have a float 3⅛ inches diameter and 2⅛ inches thick. The NA-S4 and NA-S5 models have a float of this same diameter but 1¾ inches thick. The NA-ZD5 has a float of the same diameter but 2⅝ inches thick. The NA-S6, NA-S7, NA-U5, NA-U6 and NA-V6 models have a float 3⅝ inches diameter, 1$^{15}/_{16}$ inches thick. The NA-D6 model has a spherical float 3$^{5}/_{16}$ inches diameter, which does not fit any of the other carburetor models. The NA-Y5, NA-Y6, NA-Y7 and NA-S12 models all have special float constructions which will fit no other models.

The Fuel Jet Systems.—The action of fuel delivery in these carburetors may be considered under two heads, the "Main Jet" and the "Idling Jet" systems. As previously explained, the Main Jet system controls the fuel feed in the upper half of the engine speed range, while the Idling Jet system controls the lower half, though on quick opening of the throttle at low speed the main jet comes into action. For instance, if the carburetor were properly fitted to an engine turning a propeller at a maximum speed of 1,800 revolutions per minute, the main jet system would supply the fuel throughout the range from 1,800 r.p.m. down to 900 r.p.m. (as the power required corresponds to the cube of the speed, the power needed to turn the airscrew at 900 r.p.m. would be only one-eighth that needed at full speed). From 900 r.p.m. down to idling, the fuel is delivered and the mixture proportion controlled by the idling jet system. When the engine is idling, very little throttle opening is required and the vacuum in the intake manifold is relatively high. If the throttle be opened suddenly during these

low speeds, air will be drawn in not only to supply whatever cylinder may be on its suction stroke, but also to first fill the intake manifold. This quick rush of air is temporarily very strong and brings the main jet into operation. The engine speed increases due to the throttle opening and the main jet continues to function.

The Main Jet System.—This involves—A "Metering Jet" at the bottom through which all the fuel is drawn from the float chamber and which regulates the high speed mixture proportion. A "Main Discharge Assembly," which includes:—

(a) A small air passage and "air bleed" holes through which air is drawn into the main channel passage to be discharged along with the gasoline.

(b) An emulsion passage and a number of discharge holes located so as to spray this fuel emulsion evenly through the column of entering air.

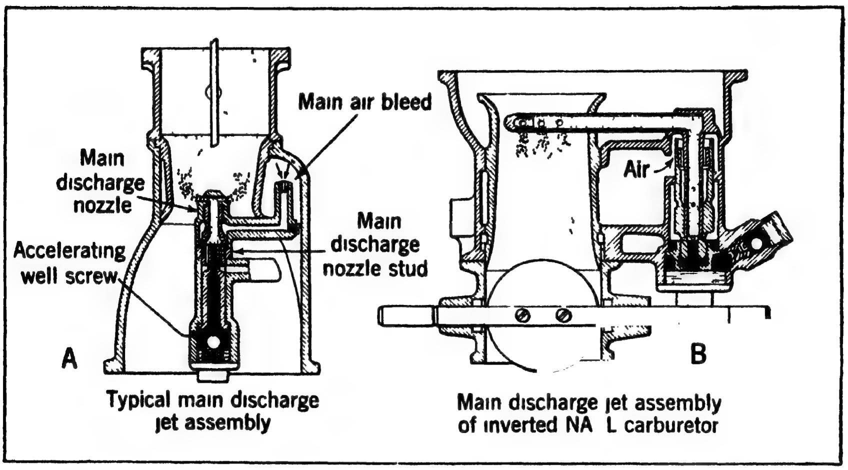


Fig. 126.—Typical Main Discharge Jet Assembly of Stromberg Carburetor at A. Main Discharge Jet of Inverted NA-L Type Carburetor at B.

(c) An accelerating well chamber around the main channel passage, which holds a reserve supply of fuel to be discharged as the throttle is opened, thereby insuring a positive response and pickup of the engine without stalling, missing or backfiring.

(d) Openings which conduct to the idling metering orifice.

The metering jets are numbered to indicate their size, the numbers ranging according to the Twist Drill and Steel Wire Gauge. Details of construction, tables of size, flow capacity and methods of testing are given later.

The Main Discharge Assembly.—The body of the discharge nozzle is usually made an aluminum casting, which carries the air bleed opening and emulsion discharge holes. A central bushing or stud of brass is screwed very tightly into this casting and the two are held in the carburetor by the brass accelerating well screw, as shown in Fig. 126 A.

The bore of the main discharge nozzle stud forms the passage through which the emulsion of fuel and air travels and should bear a certain definite

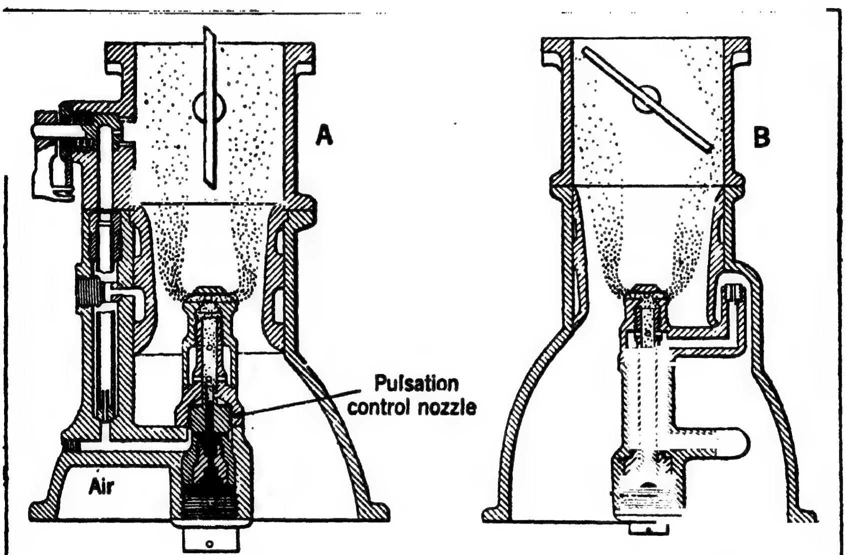
relation to the amount of fuel and air which must pass through it. If this passage is too large, the air bubbles will not fill it to give a homogeneous emulsion; if too small, the resistance to flow at high speeds will retard the fuel of this emulsion more than it does the air, resulting in a tendency toward lean mixture at high speeds. In the inverted carburetors, the main discharge jet has been made of different form, as shown in Fig. 126 B. To prevent any rise of level in the float chamber from flooding gasoline into the intake passages and thence into the engine cylinders, the wall of the main discharge jet has been made in the form of an overflow cup. The edges of this cup are beneath the cross passage to the main discharge openings and when a rise of level occurs, the fuel escapes through the air bleed restrictions and over the edge of the cup to a drain chamber and a drain pipe terminating underneath the fuselage. The remainder of the jet functions are the same as in the standard form of carburetor.

In the tubular wall of the emulsion passage are drilled a number of holes, which connect the emulsion passage with the accelerating well chamber. The upper holes are so near the fuel level that they are uncovered at the lowest suctions that will draw fuel from the main jet, so that they admit air to the emulsion passage all the time the main jet is in operation. The next lower holes into the emulsion passage furnish the desired area of opening for the fuel to pass from the accelerating well chamber to the emulsion passage for acceleration. They are usually made of aggregate area equal to or greater than the emulsion passage so that when the throttle is opened, the accelerating well may discharge as promptly as possible. The change of the accelerating well fuel level at different throttle positions is illustrated in Figs. 122 and 123.

Volume of Accelerating Well.—The volume of the accelerating well chamber has been worked out experimentally on the different carburetor models so that the excess volume of gasoline discharged when the throttle is opened, will give smooth and positive acceleration with the engine cold, as when gliding down from altitude. When the engine is very warm, this may give so rich a mixture as to show a slight stumble or hesitation, with some black smoke in the exhaust, when the throttle is opened quickly; as this rich mixture only lasts for two or three suction strokes, there is no danger of plugs fouling or the engine stalling from this cause, while the advantages of the rich accelerating charge for a cold engine are obvious. The discharge of black smoke may be cut down by reducing the size or number of the second row of holes in the emulsion passage and also by filling up part of the space in the accelerating well chamber with a bushing or plug. The size of the holes in the sides of the emulsion passage to a certain extent determines the size of the air bubbles formed inside the jet and it has been found that the smaller size holes give a finer emulsion and a steadier flow. For this reason a number of small holes are used rather than fewer large ones. At the bottom of the emulsion passage are located what are known as the idle feed holes, through which the fuel is drawn to the idling metering jet and idling system during closed throttle operation. The aggregate area of these holes is made considerably larger than that of the idling metering jet so that they cause no restriction. At full open throttle and full speed, air comes back into the main discharge passage from the

idling system, and this has a tendency to make the mixture leaner than if such air flow or air bleed were not taking place.

This tendency is least when the air from this source comes in with the main air bleed at about the level of the lower rim of accelerating well feed holes, as shown in Fig. 126 A which illustrates the construction used when it is desired that the part throttle or cruising mixture range shall be as lean or leaner than the full power range. Due to the effect of pulsations, as previously explained, with three or fewer cylinders pulling from one carburetor air opening, the suction is higher in proportion to the air flow at wide open throttle than at part throttle, so that from this cause there may be a tendency to give a too rich mixture at full open throttle if the metering jet size has been selected to give the correct mixture for part throttle position. The leaning out effect of the reverse air bleed through the idling channel at wide



Main discharge jet of NA-S4 carburetor on Wright-Lawrence radial engine, at full open throttle. Note air flow from idling system to create emulsion condition in reduced passage of pulsation control nozzle.

Main discharge jet of NA-S6a carburetor used on U. S. A. W-1 engine. Note idle feed taken from lower part of accelerating well space.

Fig. 127.—Main Discharge Jet of NA-S4 Stromberg Carburetor at A. Main Discharge Jet of NA-S6 Carburetor at B.

open throttle may be utilized to counteract this pulsation effect as shown in Fig. 127 A, which illustrates the main discharge system of the model NA-S4 carburetor feeding three cylinders of the Wright-Lawrance Radial air-cooled engine. In this jet form, the channel carrying the idle feed holes is made small in diameter, which augments the "leaning out" effect of the reverse idle air bleed. It is found, if the correct size metering jet be selected for partly open throttle, say from 1,400 to 1,750 r.p.m. with a propeller which permits a maximum engine speed of 1,800 r.p.m., that, without dis-

turbing the mixture setting in the part throttle range, the mixture at 1,800 can be varied by proper selection of the size of this channel. The smaller the channel is made the leaner is the mixture at full speed. When of this form, the accelerating well screw is customarily designated as the "pulsation control nozzle."

An alternative arrangement of the idling feed holes is shown in Fig. 127 B, which illustrates the main discharge jet used in the NA-S6A carburetors on the U. S. A. W-1 engines. In this carburetor, the fuel supply is drawn from the accelerating well space and the idling feed holes have to be somewhat larger than in other constructions, due to the fact that the flow through them takes place only by gravity and not by suction. If these holes are too small, there may be a lean spot in the range at 600 to 1,000 r.p.m. which cannot be cured by changing either the idling adjustment, idle air bleed or metering jet size.

The Idling Jet System.—As previously explained, during the lower half of the engine speed range the suction on the main discharge jet is very low, scarcely sufficient to lift fuel from it. At the same time, however, there is a very high suction on the intake manifold side of the throttle and the fuel feed is, therefore, arranged to deliver into this region of high suction. As shown in Fig. 128 A, to effect this there is a complete discharge jet system in miniature, with fuel metering jet, air bleed and discharge jet, opening into the small air passage around the throttle formed by the slot in the idling discharge jet.

The method of mixture graduation of this small carburetor element is entirely different from that of the large one for the reason that the greatest suction in the intake manifold above the throttle is at the lowest speed, when the least air is being taken in, and consequently when the least fuel is needed. As the engine speed increases and more fuel is needed, the suction in the manifold diminishes. To overcome this difficulty, the idling fuel system does not meter under the suction existing in the intake manifold, but instead is controlled by the suction existing in a small intermediate chamber or slot, situated in the wall of the carburetor at the edge of the throttle valve and having openings in the barrel both above and below the throttle. Fig. 128 A illustrates the construction used, "N" indicating the position of the throttle valve at 400 r.p.m. (propeller load) and "M" the position at 600 r.p.m., the air opening around the throttle in the carburetor barrel, required by the increased engine speed, being approximately 50 per cent greater at "M" than "N." When the throttle is in the position "N," the opening from the idling slot to the venturi tube is considerably larger than that from the idling slot to the intake manifold, and the suction in the idling slot is more nearly the low suction existing in the venturi than the high suction in the intake manifold. When the throttle has taken the position "M," for 600 r.p.m. of the engine, the suction in the idling slot is higher than before, because the openings to the intake manifold and to the venturi are about equal and the suction in the slot is about half way between those above and below it. At 900 r.p.m., the throttle will have opened still further, so that all of the slot is in communication with the intake manifold, and even though the suction in the intake manifold is lower than it was at lower speeds, the suction in the slot is higher. By having this increase in

suction in the idling slot properly graduated, the correct fuel feed can be maintained throughout the low speed range.

Idling Adjustment.—While the throttle edge is passing the idling slot, the mixture is controlled by the amount of slot opening above and below the throttle edge, in conjunction with the set size of idle metering jet and

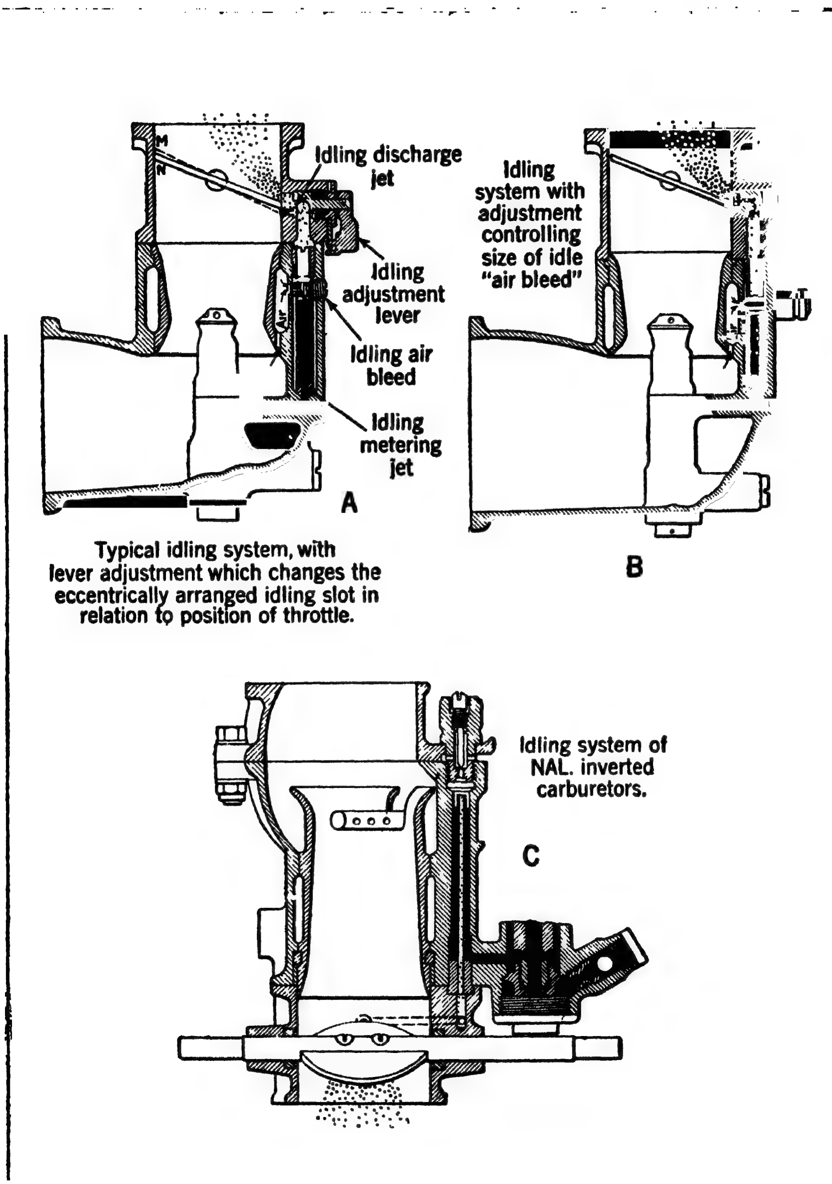


Fig. 128.—Typical Stromberg Carburetor Idling System at A. Idling System with Adjustment Controlling Size of Idle Air Bleed at B. Idling System of NA-L Inverted Carburetor at C.

the idle air bleed, and any of these may be varied to give the adjustment. On the model NA-L5, NA-S6 and NA-U6 carburetors, the idle adjustment is obtained by controlling the size of the idling air bleed with a needle point valve adjustment (see Fig. 128 B); screwing the needle inward gives less air and a richer mixture, outward the opposite. On later models the idling air bleed is made of fixed size and the idle adjustment obtained by rotating the idling discharge slot to expose more or less area above the throttle edge, there being a quadrant on the outside of the carburetor to show the range of adjustment. The first mentioned adjustment, acting on the air bleed, affects the whole low speed range about equally—from the lowest idling to about 900 r.p.m. With the later or lever type the effect of the adjustment is the strongest at very low speeds and disappears entirely at 600 to 700 r.p.m. When the lever type is used a convenient means of setting for speeds from 650 to 900, without affecting either the low idle or high speed, is afforded by selection of the proper idling air bleed size, a larger bleeder making the mixture leaner, a smaller one richer.

The idle metering orifice is usually located in the lower end of the idle tube but in some models is in a separate part or jet which is screwed into place. The tube or jet must seat securely, otherwise the effect of a larger idle metering orifice will be obtained, resulting in a richer mixture at low engine speeds and a leaner mixture, due to the larger reverse air bleed into the main jet, at full speed. The idle metering orifice is usually made about .20 to .25 of the area of the main metering jet. An exact selection of the size is unnecessary on account of the range of adjustment available by other means. The requirements of the idling system on the inverted carburetors are peculiar in that the fuel, after being drawn from the lower part of the main discharge jet passage, must be raised to a point above the fuel level and then conducted downward to the idle discharge jet at the edge of the throttle (see Fig. 128 C); if this were not done, the fuel could drain continuously from the float chamber through the idling jet into the intake manifold with the engine not running. As shown in the illustration, the idling fuel rises around the exterior of the idling tube and then goes down its interior to the discharge jet, the idling air bleed being located at the uppermost part of the passage where it will prevent any syphoning action. In the model NA-L5A carburetor, used on the supercharged Liberty engine, the idle discharge jet is made with a special shape, also illustrated while the throttle edges adjacent to the idling discharge jet have been reduced to $\frac{1}{16}$-inch thickness. On these carburetors the idling air bleed constitutes the adjustment.

The Altitude Mixture Control.—As the airplane ascends in altitude the atmosphere decreases in pressure, temperature and density. The weight of each air charge taken into the engine decreases with the decrease in air density, cutting down the power in about the same percentage. In addition the mixture proportion delivered by the carburetor is affected, the mixture becoming richer at a rate inversely proportional to the square root of change in air density. In the Stromberg line of airplane carburetors, two different methods, the Float Chamber Suction Control and the Air Port Control as shown at Fig. 129, have been employed for correcting this tendency toward enrichment of the mixture with increasing altitude. Both of

these, however, operate by decreasing the suction tending to draw the fuel through the metering jet. The float chamber suction type of control operates to reduce the fuel flow by placing a certain proportion of the air passage suction upon the fuel in the float chamber so that it opposes the suction existing in the main discharge jet. The air port control operates by bypassing part of the air charge around the venturi tube and fuel jet, and reducing

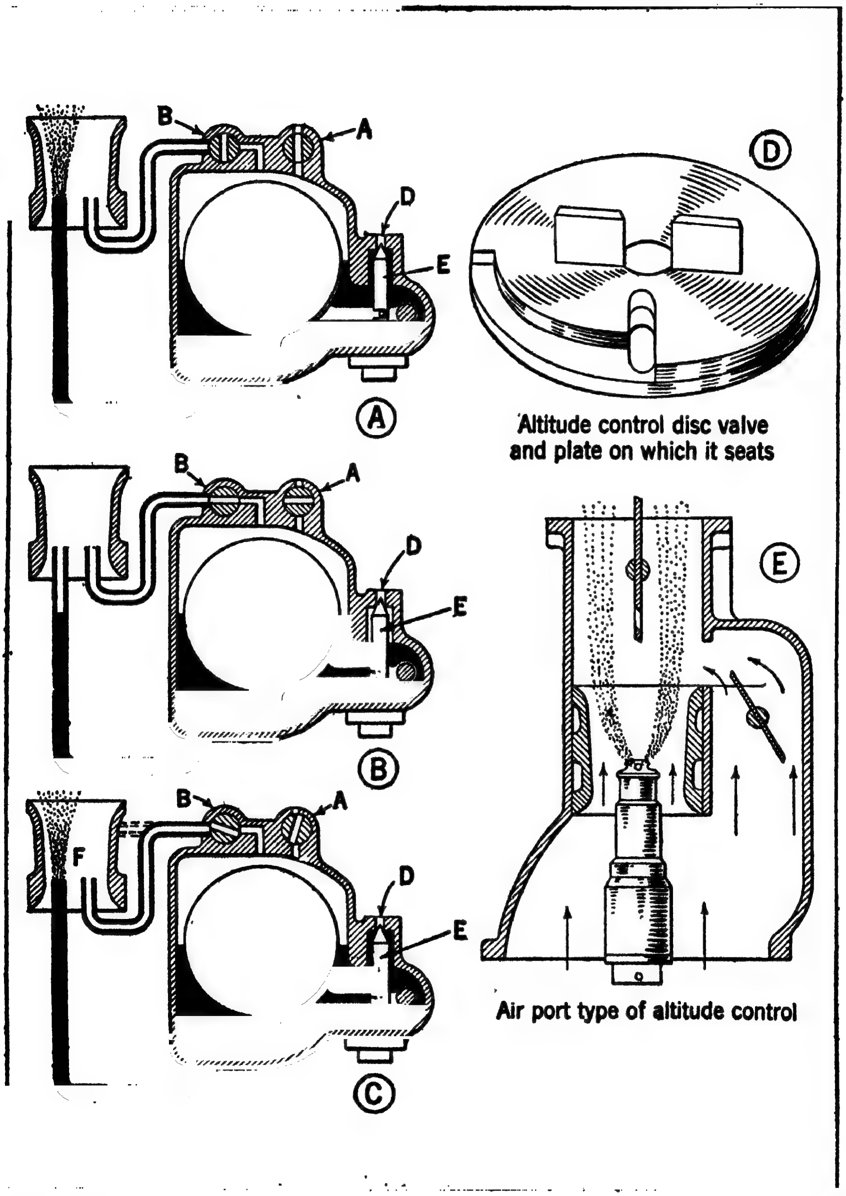


Fig. 129.—Float Chamber Control of Stromberg Aircraft Carburetors at A, B, C. Altitude Control Disc Valve Shown at D. Air Port Type of Altitude Control Shown at E.

both the air velocity and suction existing at the main discharge jet. The air port type of control has the advantage that it slightly increases the air volume capacity of the carburetor with increasing altitude; but the allowable reduction of velocity past the jet reaches a limit at about 20,000 feet and this type of control is, therefore, not suitable for extremely high altitude service. The air port control also requires an increase in carburetor size which often prevents its use when space is limited, as, for instance, in the Vee of a twelve-cylinder engine.

Altitude Mixture Control Range.—As previously stated, the air port altitude controls are designed with a range of correction to about 20,000 feet, which means that with the control in the full lean position, the fuel flow is equivalent to that which would be obtained by the use of no control but a fuel jet of 28 per cent less area. Similarly the float chamber suction type of control is made with a correction range of 25,000 feet, which is the equivalent of reducing the jet size 36 per cent. It will be obvious that the whole of this correction will only be available if the jet size is correct for ground operation. If a metering jet setting is selected which gives a mixture ten per cent richer than necessary on the ground, relying upon the altitude control to obtain the proper ground and low flying setting, obviously the remaining correction available for altitude use will be less than if the ground setting were obtained with a smaller jet and the altitude control full rich. The term "limit of altitude correction" refers to the maintenance of the "best setting," or leanest mixture of maximum power. After the limit of altitude correction has been reached, the plane can ascend 6,000 to 10,000 feet higher before the mixture will become so rich as to actually cause the motor to lose power, although the fuel consumption will become unnecessarily high and the engine may run somewhat roughly.

Float Chamber Suction Control.—The method by which this operates may be understood by consideration of Fig. 129 A, B and C. This simple carburetor has an air entrance, a fuel nozzle and a float chamber fuel supply with two openings at the top, one connected to the same suction as the fuel nozzle and the other connected to a region of no suction, these connections having valves, "B" and "A" respectively, which can be opened or closed.

In Fig. 129 A, if the valve "B" be closed and the valve "A" wide open, there is the ordinary and usual condition of carburetor action, with suction on the jet, and no suction, but simply atmospheric pressure, on the float chamber. This condition exists when the mixture control is in the full rich position. If, as shown in Fig. 129 B, the valve "B" were open and valve "A" closed, no fuel would discharge, because the suction being the same on either side of the metering jet, there would be no reason for the fuel to flow through it, and the fuel would simply take the level shown. There would, of course, be suction above the fuel in the float chamber, which would tend to draw more fuel through the needle opening "D," but provided the float were sufficiently large, the valve "E" will hold shut and maintain the level in the float chamber at the normal height. This corresponds to the extreme "lean" condition that could be obtained with this type of control, in which there would be no fuel flowing at all. In actual construction, the suction connection is taken from a location of lower suction, as shown by the dotted

lines, so that with the valve "A" entirely shut some fuel will flow. This condition usually corresponds to a correction of 30,000 feet altitude. Fig. 129 C shows how an intermediate condition may be obtained. If the cocks "A" and "B" be partially open the pressure in the float chamber will not be equal to the full suction on the jet, nor will it be atmospheric pressure, but somewhere between, depending upon the relative openings at "A" and "B." The rate of fuel discharge will consequently be between those of Fig. 129 A and B. And so long as "A" and "B" are left in one position, the pressure in the float chamber will always be the same percentage of the suction at "F" regardless of how the suction at "F" may vary, so that the action of any setting is uniform at all working speeds.

In these carburetors the desired range of control is so limited that the valve "B" may be dispensed with, a small hole of fixed size being used instead, while the total regulation is accomplished by motion of the valve "A." In order that the action of the control be not sensitive, the closure of the valve must be rapid at first and then more gradual and this is obtained by the use of a peculiarly shaped flat disc valve working upon an elongated opening or slot. (See Fig. 129 D.) The construction of the assembly is shown on Fig. 121. A spring performs the double function of holding the control valve to its seat and of producing enough friction to keep the altitude control lever from vibrating. In all new models having this type of control, the control valve is made an integral part of the carburetor, except on the model NA-L5 as used on the Liberty twelve-cylinder engine. In this case, one control valve is used for two carburetors, the valve drawing air from the rear air stack and delivering it to a manifold pipe leading to the top of the float chamber of each carburetor.

Air Port Control.—The construction of the air port or auxiliary air type of control is shown in Fig. 129 E. As can be easily seen, this type of control operates to introduce air which does not pass through the carburetor venturi tube, to the engine manifold. The air is taken from the carburetor inlet and enters the carburetor barrel just above the venturi tube. As explained elsewhere the effect of this is to lower the suction in the venturi tube. On some models a screen with vertical perforations is used to admit the air equally all around the carburetor barrel and also to permit fuel to drain down the intake passage without finding its way into the air port chamber, which might take place with low engine speeds at low altitudes when the air port valve is closed.

Combined Air Port and Float Suction Control.—On the NA-S4 carburetor for the Lawrance Radial engines, a combination of the two types of controls has been used. The float vent, which determines the pressure existing in the float chamber, is located in the air port passageway on the atmospheric side of the valve. When the valve is closed, there is substantially atmospheric pressure in the float chamber; as the valve is opened, the velocity past the fuel jet and the suction thereon are reduced and at the same time a certain degree of suction, due to the air velocity through the air port passage, is placed on the float chamber. This method combines both types of controls and gives a gradual action, increases the air capacity at high speeds, and does not reduce the velocity past the jet to undesirable

limits at high altitude. It will be noted that all of the control methods mentioned preserve the automaticity of the mixture range; that is, any given setting of the control reduces the suction on the jet by the same percentage at all engine speeds. Since the delivery of the jet bears a constant ratio to the suction, any given setting of the control has a substantially uniform effect upon the mixture at all engine speeds during which the main jet is in operation. Neither of these types of controls affects the idling jet operation.

Location of Float Chamber Atmospheric Vents.—The pressure of the propeller blast is often an appreciable percentage of the difference in pressures in the carburetor causing fuel flow and it is very important that whatever pressure disturbance is caused by the propeller blast should operate equally on both sides of the fuel metering jet, so that the fuel flow will be responsive only to the difference in pressures resulting from the flow of air through the carburetor. To insure this condition, the float vents and control openings, in all the types of controls referred to, are brought to the air entrance of the carburetor. Any pressure disturbance resulting from the propeller blast or forward motion of the ship is thereby balanced equally on the float chamber and on the fuel jet. Whatever slight depression may exist in the air entrance is transmitted to the float chamber and for this reason a manometer connected to the float chamber during dynamometer test may show some depression with either type of control in the full rich position.

The "S" Series.—The single barrel vertical carburetor is made in twelve models ranging in size from 1¼ inches to and including 2¾ inches, as follows:

NA-S4	NA-S5A	NA-S6	NA-S7	NA-S8
	NA-S5B	NA-S6B	NA-S7A	NA-S8J
		NA-S6C	NA-S7B	
		NA-S6D		

In addition there is a large single barrel model, the NA-S12.

Fig. 130 A, showing an NA-S6 carburetor is typical of these models. They are all similar in general construction, having the float chamber on one side of the barrel but differ in details. The NA-S4, the smallest made, has a combination air port and back suction type of mixture control and the lever type of idle adjustment. The NA-S5A is similar to the NA-S4 except that the mixture control is of the back suction type. The NA-S5B is identical with the NA-S5A except for the position of the throttle shaft, which is perpendicular to the axis of the float chamber instead of parallel as in the NA-S5A.

The 2¼-inch single carburetor, due to various demands, has been made in four models as listed above. The NA-S6 has the air port control and variable air bleed idle adjustment. The NA-S6B has a back suction mixture control instead of the air port. The NA-S6C differs from the NA-S6 only in having the throttle shaft at right angles to the float chamber axis instead of parallel. The NA-S6D differs from the NA-S6 in having the lever type of idle adjustment instead of the variable air bleed. The NA-S7 is a reproduction of the NA-S6 in a size one-quarter inch larger. The

NA-S7A is identical with the NA-S7 except that it has the lever type of idle adjustment. The NA-S7B has the throttle shaft at right angles to the float chamber axis but otherwise is the same as the NA-S7A. The NA-S8 is similar to the NA-S6 except that it has the back suction type of mixture control and lever type idle adjustment. The NA-S8J shown at Fig. 130 B is a special model of the NA-S8 for use primarily on air-cooled radial engines with a supercharger or rotary distributor, and has an oil-jacketed throttle half to prevent the formation of ice. Internally it is essentially the same as the NA-S8 but the throttle position is at right angles to that

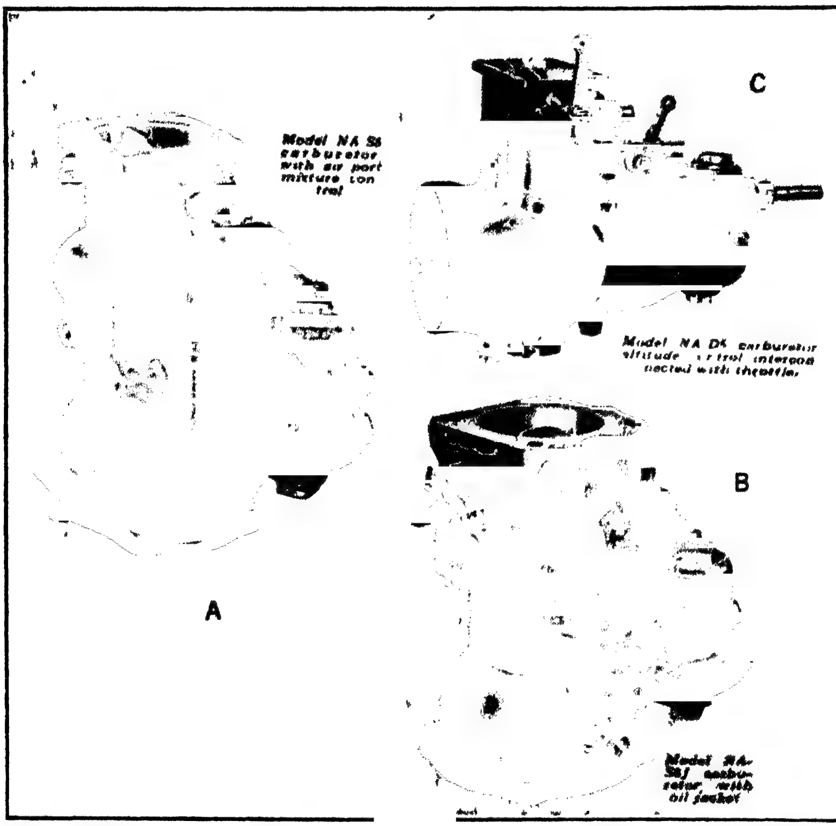


Fig. 130.—Illustration Showing Various Types of Stromberg Aircraft Engine Carburetors of the NA Series.

of the NA-S8 and the fuel inlet, idle adjustment, mixture control valve and air bleed are in different locations. The NA-S12 is a 3¼-inch carburetor with two float chambers and two discharge nozzles. The two floats, one on each side of the barrel, are connected together and operate on a single needle valve. The mixture control is the back suction type and the idle is the lever type.

Stromberg NA-R Series Carburetors.—A line of single barrel carburetors designed especially for radial air-cooled engines, has been developed by the Stromberg Motor Devices Co. These carburetors, designated as the NA-R

models, incorporate several features now in use on other Stromberg types. Included in these features are an economizer, a mechanically operated pump and a mixture control, which on the new model is of the needle valve type. Nominal barrel sizes of the new models range from 1½ inches to 3 inches and designation is in accord with standard S.A.E. practice. The lower halves of the NA-R4 and NA-R5 sizes are so arranged that either of two types of throttle half may be used, one with the throttle shaft parallel to the fore and aft centerline of the float chamber and the other with the shaft perpendicular to this centerline. Manifold flanges of the 1½ inches and 1¾ inches sizes (3 and 4) are of the S.A.E. standard two bolt type, whereas the larger sizes are of the square two bolt type. Intake flanges for the application of air cleaners or heaters are provided. Throttle and mixture controls have in most cases been designated so that the levers may be mounted in any position and overall dimensions have been made as small as possible. Float mechanisms, which are of the hinge type, are large enough to operate satisfactorily with fuel as supplied under pressure from a fuel pump or by gravity only and the float chambers have sufficient capacity and proper location of jet feed passages to insure smooth engine operation during maneuvers.

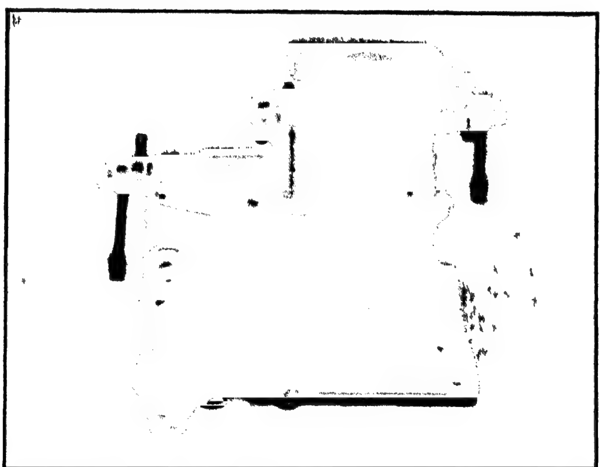


Fig. 130D.—Illustration of New Series NA-R Stromberg Aircraft Engine Carburetors.

Adjustments are provided on the throttle stop and on the idling system for regulating the quality of the mixture at idling speed. Main and economizer jets are of the fixed orifice type. The float chamber is vented to the air intake. Floats are of brass with a reinforcing strut through the center and those in the 2 inch, 2½ inch and 3 inch models are flat on the top reducing overall weight of the carburetors. The main metering system is of the plain tube type with an air bleed to the main wall. The main discharge nozzle and well construction has been simplified, one piece being used instead of the three pieces in other models. The idle metering system is of the standard Stromberg type. The economizer of the 2½ inch and 3 inch sizes consists of a needle which is held off its seat at full throttle by a forked lever fitted on one end of the throttle shaft. Directly below this

seat is a metering jet of the fixed orifice type which accurately meters the fuel passing from the float chamber through the economizer needle seat to the main discharge nozzle. As the throttle is closed the operating lever allows a spring to seat the economizer needle thus shutting off the flow of fuel through the economizer system. This permits the use of a rich mixture at full throttle and a lean mixture throughout the cruising range of speeds.

A pump operated by the throttle is provided to furnish a rich mixture for acceleration. This consists of an inverted cylinder having a stem at the upper end which is operated by the throttle. Within the cylinder is a piston mounted on a hollow stud screwed into the main body casting. The upper end of this stud is shaped like a small poppet valve with several holes in the wide face of the valve leading into the center hole. The valve seat is in the piston which is held up against the valve by a spring. The center hole of the stud connects with a passage leading to the main discharge nozzle. The whole assembly is mounted in the float chamber so that when the throttle is closed and the cylinder in top position the space within the cylinder is filled with fuel. As the throttle is opened rapidly the cylinder is moved down and the pressure of the fuel above the piston forces it down thus opening the valve so that fuel under pressure is forced out the main discharge nozzle. As the throttle is closed fuel is drawn into the cylinder through the clearance space between the piston and cylinder. This arrangement of filling the cylinder provides automatic regulation of the fuel charge depending upon the speed of throttle opening. For engines requiring a large accelerating charge for cold weather operation, a restriction or reducer may be used during warm weather operation.

The Double Models.—The double or two-barrel series of carburetors includes varied models, the "D" having the float chamber to the rear of the two barrels, which are side by side, the "U" in which the two barrels are separated by the float chamber whose center is approximately on a line between the centers of the barrels, the "Y" with the float chamber divided, part being in front and part in the rear of the two barrels which are side by side, and the "L," or inverted type, with the float chamber in the rear, in which the air and mixture flow are downward through the carburetor.

The model NA-D5 (see Fig. 130 C) has a back suction type of mixture control and an adjustable air bleed idle. The NA-ZD5 is a special dirigible carburetor whose outstanding feature is an oversize float to withstand large pressure heads. It has the back suction type of mixture control and lever idle adjustment.

The "U" series, of which the NA-U5 and NA-U6T, as shown in Fig. 131 A and B, are typical, includes the NA-U5, NA-U6, NA-U6A and NA-U6T. The NA-U5 and NA-U6T have the back suction type of mixture control and lever type idle adjustment. The NA-U6 has the air port type of control and variable air bleed type of idle adjustment. The NA-U6A is in general the same as the NA-U6 except that it embodies an automatic mixture control of the back suction type operated by an aneroid.

A few special models of the NA-U67 called the NA-UT3 have been built.

They are the same as the NA-U6T except that the barrel is ⅛ inch larger or 2$\frac{9}{16}$ inches in diameter.

The "Y" model (see Fig. 131 D) is designed to be used where space, and especially width, is limited and still retain the advantages of the "U" type carburetor as regards float chamber action. The float chamber is divided, part being in front and part in rear and each section has a small float. The

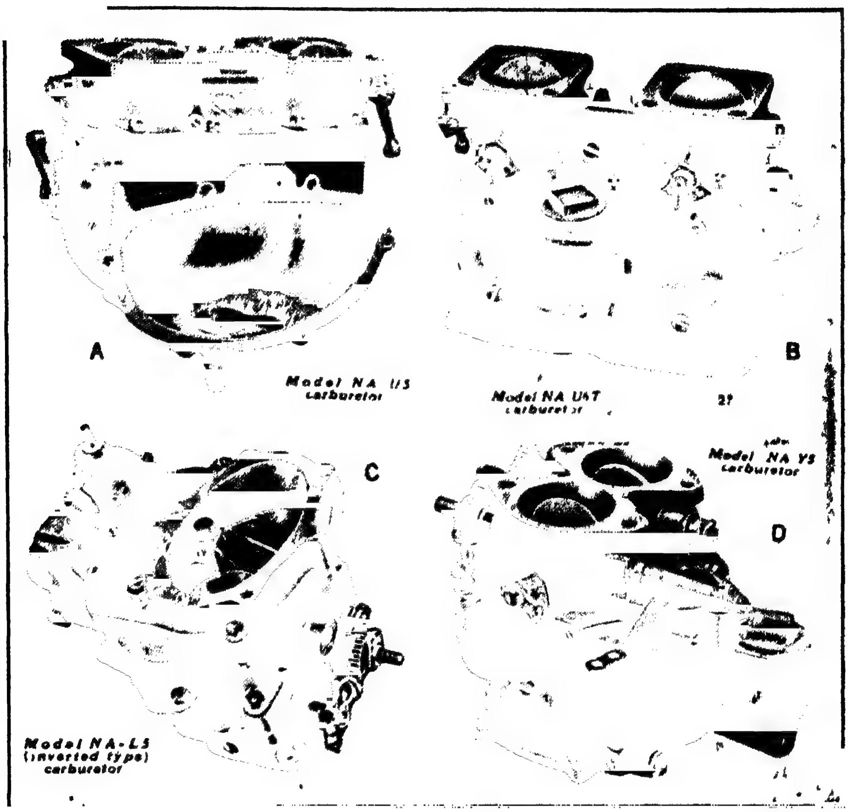


Fig. 131.—Illustration Showing Various Types of Stromberg Double Carburetors for Aircraft Engines.

floats, however, are rigidly connected together so that their action is that of a single unit. The float chamber bowls are connected at the bottom by a small tube. The model NA-Y5 has a back suction type of mixture control and a lever type idle adjustment. The NA-Y5A is the same as the NA-Y5 except that the mixture control is more powerful. The NA-Y6 and NA-Y7 are reproductions of the NA-Y5 in larger sizes. The NA-Y5, NA-Y5A and NA-Y6 have throttle shafts running from front to rear and geared together while the NA-Y7 has a single throttle shaft running from side to side.

The "L," or inverted type, built only in the two-inch size, differs in construction from all other models. (See Fig. 131 C.) Special adaptations of wells, idle tubes, discharge nozzles, etc., have been required. The float chamber is in the rear of the two barrels which are side by side. The car-

buretors provide a mounting pad on the float chamber for a mixture control valve which operates on the back suction principle. Where two carburetors are used per engine, only one valve is necessary for the two carburetors, the float chambers being balanced on each other. The idle is of the adjustable air bleed type.

QUESTIONS FOR REVIEW

1. What are the requirements for a good firing mixture?
2. What is the effect of valve overlap on mixture?
3. What causes a "blowback" in carburetor?
4. How does a Venturi tube work?
5. Why must different carburetors be used on engines of different make?
6. Describe air bleed principle of Stromberg aircraft carburetors.
7. What is the Stromberg accelerating system?
8. What is the influence of various flying maneuvers on Stromberg float action?
9. What is the difference between a "main" jet and an "idling" jet?
10. Describe Stromberg idling jet system.
11. How is altitude mixture control secured on Stromberg carburetors?
12. What is the difference between float chamber suction and air port control?
13. Why are double and triple outlets necessary?
14. Why are float chamber atmospheric vents used?
15. What is the difference between a Stromberg NAS6 and a NAU6?

CHAPTER XI

CORRECT SETTING FOR STROMBERG CARBURETORS INTAKE MANIFOLDS—AIR HEATERS—FUEL FILTERS

Determination of Carburetor Setting—Determining Venturi Size—Determining Main Metering Jet Size—Changing Accelerating Well Bore—Carburetor Settings—Installation of Stromberg Carburetors—Installing on Engine—Starting Procedure—Routine Inspection in Airplane—Complete Inspection and Overhaul—Bench Inspection—Construction of Metering Jets—Calibration of Metering Jets—Intake Manifold Design and Construction—Compensating for Various Atmospheric Temperatures—The Wright J 5 Air Stove—Installation of Pratt and Whitney "Wasp" Mixture Heaters—Utility of Fuel Strainers.

Determination of Carburetor Setting.—The determination of a fully correct carburetor setting and specification involves the use of both a dynamometer setup and an airplane installation. With the dynamometer it should be possible to vary the speed and load as desired, and provision should be made for obtaining accurate readings of power, fuel consumption and manifold vacuum. A carburetor setting is never considered as definitely correct until it has been thoroughly checked by airplane operation. It has usually been found that any discrepancy between the dynamometer and airplane performance occurs at the lower speeds. Where a dynamometer is not available, a propeller or torque stand can be used as a substitute, but the setting obtained should not be considered as final.

In obtaining a setting and specification, the first step is to determine the air capacity necessary for maximum power (or for power desired in case less than maximum is to be used), by selection of the proper size venturi tube. Following this, the proper size metering jet should be ascertained. The idle adjustment should then be made at the lowest speeds, after which the idle intermediate range should be made correct. In exceptional cases it may be found necessary to change the accelerating well bore or the main jet air bleed from those initially furnished by the factory. These changes should not be made unless found to be necessary after the venturi and main jet size have been determined and fuel consumption readings obtained over the whole running range.

A preliminary estimate of the venturi size may be made as follows: With one carburetor feeding three or fewer cylinders, the area of the space around the discharge jet at the venturi throat should equal the piston displacement in cubic inches of one cylinder *times* the approximate maximum revolutions per minute *divided by* 133,000. With one carburetor feeding four or more cylinders, the area around the fuel discharge jet at the venturi throat should equal the piston displacement of all cylinders *times* the approximate maximum revolutions per minute *divided by* 480,000. The average main metering jet orifice *diameter* is about .05 that of the venturi *diameter*. In making the dynamometer setup, the same care should be taken to see that the installation is correctly made as for an airplane installation.

Determining Venturi Size.—The venturi tube should be large enough to give the engine full power but there is no advantage in having it larger

than this. The restricting influence of the venturi tube on the air passing through it may be measured by the suction existing in the manifold just above the carburetor with throttle full open. This is the manifold vacuum and should be carefully distinguished from the carburetor vacuum which is the vacuum existing at the throat of the venturi. With four or more cylinders pulling, maximum power will usually be obtained when the mani-

STROMBERG AIRCRAFT CARBURETORS

Model	No. of Bbls.	Bbl. Dia., In.	Type of Flange	Range of Venturi Sizes Possible, Dia., Inches	Mixture Control Type	Idle Adjust. Type	Throttle Shaft Direction with Float Chamber Axis	Air Intake Flange Shape	Carburetor Weight, Pounds
NA-S4	1	1 13/16	4-bolt S. A. E.	1⅛-1⅝	Air Port and Back Suction	Lever	Parallel	Elliptical	6.3
NA-S5A	1	2 3/16	4-bolt S. A. E.	1¼-1⅞	Back Suction	Lever	Parallel	Round	7.0
NA-S5B	1	2 3/16	4-bolt S. A. E.	1¼-1⅞	Back Suction	Lever	Perpendicular	Round	7.0
NA-S6	1	2 7/16	4-bolt S. A. E.	1⅜-2 1/16	Air Port	Air Bleed	Parallel	Elliptical	7.7
NA-S6B	1	2 7/16	4-bolt S. A. E.	1⅜-2 1/16	Back Suction	Air Bleed	Parallel	Round	7.3
NA-S6C	1	2 7/16	4-bolt S. A. E.	1⅜-2 1/16	Air Port	Air Bleed	Perpendicular	Elliptical	7.7
NA-S6D	1	2 7/16	4-bolt S. A. E.	1⅜-1 1/16	Air Port	Lever	Parallel	Elliptical	7.7
NA-S7	1	2 11/16	4-bolt S. A. E.	1⅜-2¼	Air Port	Air Bleed	Parallel	Elliptical	8.3
NA-S7A	1	2 11/16	4-bolt S. A. E.	1⅜-2¼	Air Port	Lever	Parallel	Elliptical	8.3
NA-S7B	1	2 11/16	4-bolt S. A. E.	1⅜-2¼	Air Port	Lever	Perpendicular	Elliptical	8.3
NA-S8	1	2 13/16	4-bolt S. A. E.	1⅝-2½	Back Suction	Lever	Parallel	Round	7.8
NA-S8J	1	2 13/16	4-bolt S. A. E. with Studs	1⅝-2½	Back Suction	Lever	Perpendicular	Round	8.9
NA-S12	1	3 13/16	4-bolt Special	2⅜-3 3/16	Back Suction	Lever	Perpendicular both bowls	Irregular	11.0
NA-D5	2	2 3/16	6-bolt Special	1¼-1⅞	Back Suction	Air Bleed	Perpendicular	Round	9.0
NA-ZD5	2	2 3/16	6-bolt Special	1¼-1⅞	Back Suction	Lever	Perpendicular	Round	8.1
NA-U5	2	2 3/16	2-4-bolt S. A. E.	1¼-1⅞	Back Suction	Lever	Perpendicular	*Irregular	8.8
NA-L5	2	2 3/16	6-bolt Special	1¼-1⅞	Back Suction	Air Bleed	Perpendicular	Irregular	9.4
NA-Y5	2	2 3/16	4-bolt Special	1¼-1⅞	Back Suction	Lever	Perpendicular both bowls	Irregular	8.4
NA-Y5A	2	2 3/16	4-bolt Special	1¼-1⅞	Back Suction	Lever	Perpendicular both bowls	Irregular	8.4
NA-Y6	2	2 7/16	4-bolt Special	1⅜-2 1/16	Back Suction	Air Bleed	Perpendicular both bowls	Irregular	9.8
NA-Y7	2	2 11/16	6-bolt Special	1⅜-2¼	Back Suction	Lever	Parallel both bowls	Irregular	9.5
NA-U6	2	2 7/16	2-4-bolt Special	1⅜-2 1/16	Air Port	Air Bleed	Perpendicular	Irregular	13.4
NA-U6A	2	2 7/16	2-4-bolt Special	1⅜-2 1/16	Auto. Back Suction	Air Bleed	Perpendicular	Irregular	14.5
NA-U6T	2	2 7/16	2-4-bolt Special	1⅜-2 1/16	Back Suction	Lever	Perpendicular	Irregular	8.8
NA-U6T3	2	2 3/16	2-4-bolt Special	1⅜-2⅛	Back Suction	Lever	Perpendicular	Irregular	8.8

*Some carburetors of this model have been furnished with an air scoop having a circular entrance, and weighing 1.5 lb.

fold vacuum is about 1⅜ inches mercury or 15 inches water; with one carburetor feeding three cylinders, the maximum manifold suction for maximum power will be about ⅞ inch mercury or 11½ inches water. In making these measurements, it should be borne in mind that closing the throttle will greatly increase the manifold suction and a suction gauge, if of the "U" tube type, should be protected against emptying into the manifold when the throttle is so closed. While checking the venturi tube size, a fuel metering jet that is known to be adequately large should be used and the best mixture setting obtained with the altitude mixture control.

Determining Main Metering Jet Size.—Experience has shown that satisfactory results are obtained when the engine is tested with the relative

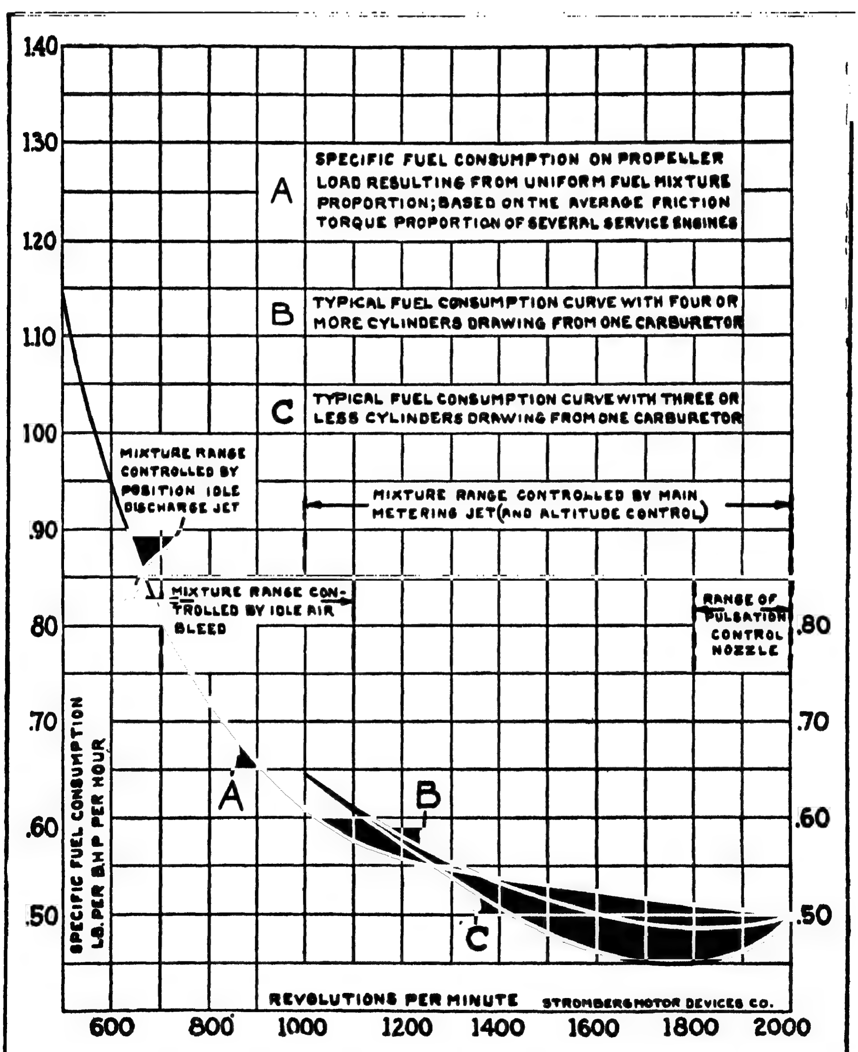


Fig. 132.—Typical Fuel Consumption Curves of Stromberg Aircraft Carburetors.

speeds and loads of propeller load; that is, starting from the contemplated normal speed at full load, the torque should be decreased as the square of the speed. With an engine of average mechanical efficiency, a uniform mixture of fuel and air would give the specific fuel consumption of the curve "A," Fig. 132. Actually, advantage is taken of the fact that the mixture furnished at part throttle can be much leaner than that required for maximum power and curves of fuel consumption as "B" and "C" are obtained. Different points on the mixture range are controlled by different elements of the carburetor as shown in Fig. 132. From about half speed up, the mixture proportion is controlled mainly by the size of the metering jet and, of course, by the altitude mixture control. Below half speed the mixture is controlled by the idling air bleed and at extreme idling by the setting of the idling discharge jet, according to the amount of its semicircular slot which shows above the throttle. On earlier models, as previously explained, the idling discharge jet was set at the Stromberg factory and the idling adjustment accomplished by a taper needle valve working on the idling air bleed. On most of the later models, the idling air bleed is set by plugs drilled in different sizes and the lower idling adjustment accomplished by rotating the idling discharge jet to bring more or less of its semicircular opening above the throttle. To find the proper size metering jet, reduce the jet size until the power begins tc drop off with indications of a lean mixture. The proper size is the next larger one to that which showed the first slight drop in power. When the proper jet size is obtained, the engine will be fairly sensitive to the use of the altitude mixture control, beginning to show lean and lose power with a small motion of the control lever and this can be used as a check.

The throttle or throttles should next be closed to give the minimum idling speed and the idling adjustment made. At this time care should be taken that the throttle openings are synchronized to the extent that the exhausts of the cylinders fed by the different carburetors are of approximately equal temperature. Care should also be taken that there are no leaks in the intake manifold system either at the flange connections or at packed telescoping joints. A fairly rich idling mixture is usually desired. After the idling adjustment is made, the throttle should be opened to give an engine speed under propeller load about 100 revolutions less than half the normal maximum speed. This is just below the point the main discharge jet comes into action and the mixture is best controlled here by change of the idling air bleed, a smaller air bleeder making the mixture richer and a larger one leaner. The setting should be such as to give a fairly rich mixture as this is the speed usually used in the field for warming up the engine. On the variable air bleed type of idling adjustment the air bleed size is not fixed and the action in this range will have to be set as a compromise with the action at minimum idling speed. With the normal type accelerating well nozzle, change in the idling adjustment or idling air bleed should not affect the mixture range very far above half engine speed.

When three or fewer cylinders draw from one carburetor barrel, there are intervals of time during which no intake valves are open, resulting in strongly marked pulsations of suction. Under these conditions the mixture

tends to be richer at wide open throttle and full speed than when the throttle is partly closed as the closed throttle somewhat dampens the effect of the pulsations as transmitted to the carburetor jets. A mixture curve, therefore, tends to differ from the curve given by the same carburetor on a four cylinder engine, in that it grows perceptibly rich at full throttle, as shown in Fig. 132. As a means of regulating this amount of richness, the Pulsation Control Accelerating well nozzle is used. With this, advantage is taken of the fact that at wide open throttle the flow through the idling system reverses and instead of fuel going through the idling nozzle as at low speeds, air is drawn back through the idling air bleed and through the idling metering jet into the main jet system. By using a reduced opening in the pulsation control accelerating well nozzle, this action of air flowing in the jet is made to give a leaner mixture than when the air does not so flow, the extent of leanness depending upon the amount the pulsation control nozzle is reduced in size.

When the pulsation control nozzle is used, a metering jet size to give the correct mixture from half speed up to within 100 revolutions of full load should be determined. At full load the desired mixture should be obtained by selection of the proper size pulsation control accelerating well nozzle. The low speed idling adjustment and idling air bleed adjustment should then be made as described above. A very considerable change in the size of the idling air bleed may have some slight effect upon the pulsation control nozzle size necessary.

Changing Accelerating Well Bore.—It may sometimes occur that when a considerably larger venturi is placed in the carburetor than that which was sent from the factory, and the proper main metering jet size determined for full speed, as the speed is decreased four to six hundred revolutions, the mixture becomes rich. This may well be shown by the fact that the mixture can be made much leaner by use of the altitude mixture control without affecting the engine operation. This is an indication that the bore of the accelerating well is too small and should be enlarged about in the same proportion as was the venturi area. It will be noted that for structural reasons the fuel passage in the main discharge system is made in two parts and the upper bore is the accelerating well bore. Enlarging the accelerating well bore will permit the use of a smaller metering jet and will bring the upper half of the speed range more nearly uniform as to mixture.

In general, decreasing the size of the main air bleed has the same effect as enlarging the accelerating well bore but the action is much less pronounced and such large changes are necessary that it is preferable to work with the accelerating well bore.

Carburetor Setting.—Below are listed some typical carburetor settings that have been used on some of the more common engines. It should be understood that these are not furnished as the correct settings which are given in the individual engine instruction books. However, where the correct setting is not known and it is desired to make a carburetor installation, they may be used as temporary expedients and will probably give satisfactory engine operation.

Installation of Stromberg Carburetors.—There are many factors involved in proper carburetor operation in service which cannot be cared for in the carburetor design alone. A recognition of the requirements regarding tanks, fuel lines, freezing in the manifolds above the carburetors and the effect of rapid acceleration, including catapulting, must be had if the carburetor installation in an airplane is to be successful. The tanks and fuel lines must be so located that there is no tendency for an air lock to form between the tanks and the carburetor and obstruct the fuel flow. Fig. 133 shows how a seemingly simple installation can contain the factors necessary for the formation of an air lock which could prevent the flow of any fuel

TYPICAL STROMBERG CARBURETOR SETTINGS

Carburetor Model	Engine	Main Metering Jet Drill Size	Venturi Tube Diameter, Inches	Main Air Bleed Drill Size	Float Level Inches Below Parting Surface	Miscellaneous
NA-S4	Wright Lawrance L-4 (3 cylinder)	49	$1\frac{3}{8}$	50	$1\frac{3}{8}$	Pulsation control nozzle
NA-S4	Wright Lawrance J-1 (9 cylinder)	47	$1\frac{7}{16}$	50	$1\frac{3}{8}$	
NA-U5	Wright E-4	47	$1\frac{5}{8}$	49	$1\frac{1}{2}$	
NA-D5	Wright E-3	44	$1\frac{19}{32}$	50	$1\frac{3}{8}$	
NA-Y5 and NA-Y5A	Curtiss D-12	45	$1\frac{11}{16}$	49	$\frac{15}{16}$	
NA-L5A	Liberty	44	$1\frac{19}{32}$		$1\frac{7}{16}$	Air bleed fixed
NA-S6 and NA-S6C	U. S. W-1A	48	$1\frac{9}{16}$	45	$1\frac{3}{4}$	
NA-U6	Wright H H-2 and H-3	42	$1\frac{13}{16}$	49	$1\frac{1}{2}$	
NA-U6T	Wright T-3	36	$2\frac{1}{16}$	49	$1\frac{1}{2}$	This is revised setting and goes with $\frac{7}{32}$-inch well bore and No. 49 idle air bleed.

from the tank to the carburetor. As is well known, the vaporizing of fuel is only accomplished through the taking up by the fuel of a considerable quantity of heat. This heat is taken from the space immediately above the carburetor. Unless heat is supplied to this space, the temperature drops to a very low value and, besides the interference with vaporization, ice will form under certain atmospheric conditions. This ice formation constitutes a serious danger. The manifolds should be well heated, particularly above the carburetor and preferably throughout their length. Hot water, exhaust gas or oil may be used for this purpose. It is often beneficial, especially for winter or cold weather operation and where the inlet manifolds of the engine are of excessive length, to heat the inlet air passing to the carburetor.

Rapid airplane accelerations, with the consequent fuel displacements and changes of level, are often greater than that of gravity and this is particularly true in the case of catapulting. The carburetors are so designed that even with these changes of level, fuel will continue to flow from the metering jets at the instant of acceleration. To continue functioning, however, the fuel supply to the carburetors must continue as normally and for this to be accomplished the locations of the tanks, fuel lines and carburetor must be so related that, with these conditions existing, the natural tendency of flow is to the carburetor and not away from it.

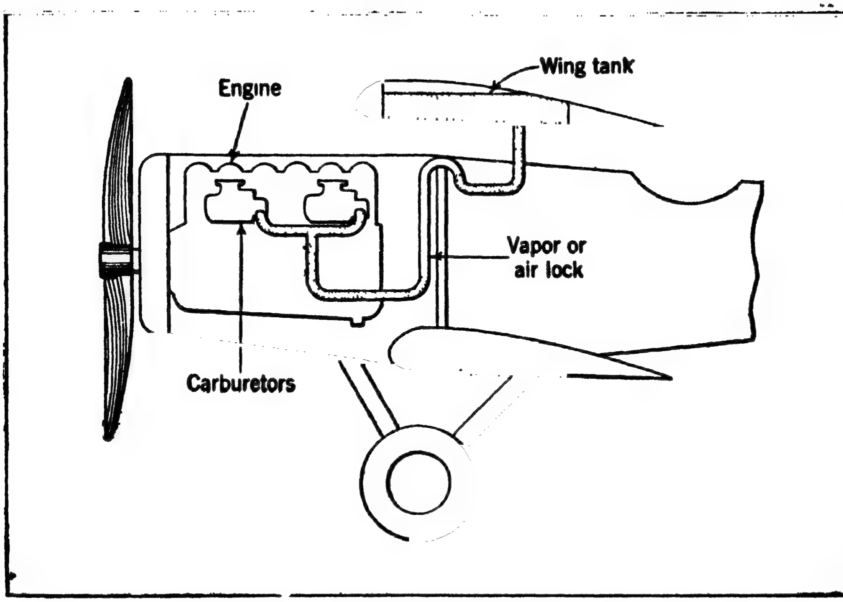


Fig. 133.—Showing How a Vapor Lock in Fuel Line from a Gravity Tank can Prevent Flow of Fuel to the Carburetor. The Height of the Fuel Column at the Left Just Balances that of Fuel Column at the Right, Giving Equilibrium Without Flow.

Installing on Engine.—Before mounting the carburetor on the engine, it should be thoroughly checked to see that all screws, plugs, bolts, etc., are "safetied" with lock washers, cotter keys or safety wire. Make certain that the proper manifold flange gaskets are in place. A typical installation is shown at Fig. 134, which shows the rear view of a Wright Whirlwind engine. This view also shows the mixture distribution and manifolding system, and incorporation of oil cooler with lower manifold.

The throttle discs should be examined closely to see that they fit securely in the barrels and if there is more than one disc to a carburetor a careful check should be made to see that they are perfectly synchronized. The action of the engine at minimum speeds depends to a large extent upon the synchronization of the throttles. The discs should never be hammered as this obviates any possibility of securing a fit. The flanges should be pulled tightly together. If more than one carburetor is used per engine, the throttles of the carburetors should be carefully synchronized with each other. This is usually done by adjusting the throttles

so that they are in exactly the same relative position at extreme closed throttle, making certain that the throttle stop has not interfered with the seating of the discs in the barrels. For an airplane installation the cockpit control rods should then be connected to the corresponding levers on the carburetors. After connection these should be carefully gone over to see that the full limit of travel is obtained for each lever and that the levers move in the proper direction as indicated in the cockpit. The fuel and drain lines before being connected to the carburetor should be flushed out to make certain that the flow is free.

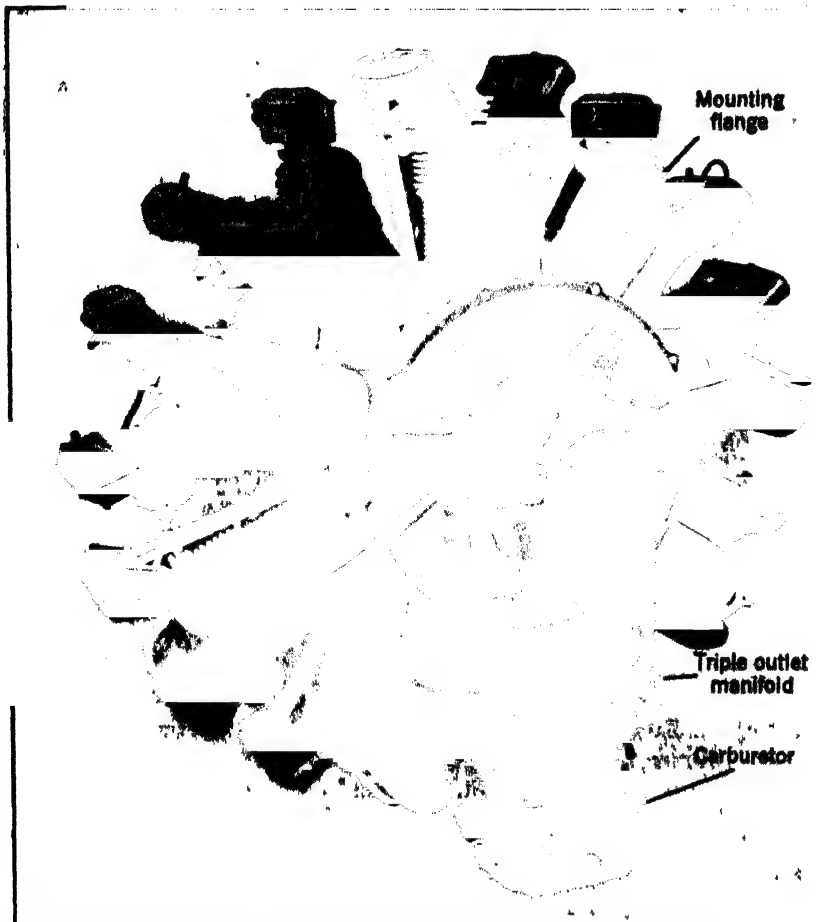


Fig. 134.—Rear View of Wright "Whirlwind" Engine, Showing Installation of Stromberg Aircraft Carburetor of the Triple Outlet Form Attached to Combined Oil Cooler and Induction Manifold.

Starting Procedure.—Before starting the engine for the first time, it is better to set the idling adjustments well toward the "rich" position to make certain that the engine secures sufficient fuel. To start the engine, close the throttle and turn the propeller in the forward or running direction through three or four complete revolutions. Open throttle slightly

and try with hand starting magneto or, if the engine is not provided with this magneto, "pull through" by hand in the usual manner. If the engine does not start, close the throttle and turn the propeller again by hand through one or two revolutions. Try the hand magneto. The wells in these carburetors are of the self-priming type and there is a possibility, especially in warm weather, of getting the engine in the so-called "loaded up" condition. Therefore, do not continue to turn the engine forward with the throttle closed if it has failed to start but open the throttle and turn the engine backward for three or four revolutions. Then close the throttle to the starting position, turn forward once or twice and try. If at any time, it is desired not to use the self-priming action of the carburetor, this can be done by opening the throttle about a third of its total movement. For starting with an electric starter, the throttle should be closed until the starter has turned the engine over through three or four revolutions, then work throttle by alternately opening to about mid position and closing, until engine starts.

After the engine has started, it should be allowed to become completely "warmed up." If this is not done, any adjustments made will be useless. When the engine is thoroughly warm the minimum speed operation should be made satisfactory. This is done by means of the idling adjustments and the throttle stop screws which regulate the minimum throttle opening. It will generally be found that the best operation is obtained with an idling mixture amply rich. Care should be taken though to see that it is not too rich so that the engine "lopes" or "loads up" when idled for a long period of time. A good indication of a too rich idling mixture is the operation of the cylinders on an eight stroke cycle; that is, they will fire only once for every two openings of the exhaust valve. The next step is to try the engine operation over the whole range by gradually opening the throttle so that the engine speed increases by increments of about 100 r.p.m. On carburetors with an adjustable air bleed idle, it may sometimes be found that there is a slight "flat spot" at around 700-900 r.p.m. This can usually be remedied by use of the idling adjustments. If this is done, the engine should be immediately brought back to idling speed to see that the operation here is still satisfactory. The acceleration should then be tried by rapidly opening the throttle from different engine speeds over the range from extreme idling to 600 or 700 r.p.m. Test the operation of the mixture control by opening the throttle and allowing the engine to come to full speed and then moving the control slowly in the "lean" direction. If the engine speed is greatly reduced, the control is functioning properly.

Routine Inspection in Airplane.—The carburetor strainer should be inspected after every flight to see that it is not clogged at any place and to clean out the accumulated foreign matter. In addition, a regular inspection should be made of the feed and drain lines to see that they have a free flow. The controls should be thoroughly checked to see that they do not stick and that there is no lost motion. The carburetor should be examined, making certain that all safety wires, cotter keys, etc., are in place and that no parts have become loose. Then "run up" the engine as given under "Starting."

Complete Inspection and Overhaul.—This should be done in any case where the exact condition of the carburetor is not known, where the carburetor has been in service for a long period of time or where the carburetor action is known to be bad. The carburetor should be completely disassembled, removing all parts including the float mechanism, but excepting the throttle valves whose condition can be observed in place. Also the idle discharge jets, on carburetors which have the air bleed idle adjustment, should not be disturbed because of the accuracy of adjustment required in replacement. If, however, these idle discharge jets have been accidentally displaced the correct setting will be approximated when one-fourth of the half moon opening shows above the throttle valve (on manifold side), when same is entirely closed.

It will be found that tapping the brass plugs lightly with a soft wood or horsehide mallet will aid greatly in their removal from the aluminum body. Care should be taken that the carburetor is not injured. Wash the carburetor body and parts in gasoline or its equivalent, using an air hose to blow through the passages and clean the different parts. Then examine *all* passages to see that they are clean. To do this the carburetor must be held to a good light or a flash light used. A good method of making this examination which insures that no passages will be overlooked is to follow the flow of the fuel from the float chamber through the drilled holes, metering jets, wells and well tubes to both the main and idling discharges. Check the air bleed passages to see that they are clean. Check the metering jet and air bleed plug sizes against the sizes marked on the aluminum setting plate. See that the idling adjustments are correctly in place and that the throttle valves fit snugly in the barrels and are well synchronized. The later double models with parallel throttle shafts geared together have a means of adjustment to obtain accurate synchronization. One of the pair of gear sectors is not pinned to the shaft, but is clamped by means of a bolt and nut. Loosening this nut will allow the gear sector to be turned on the shaft and an adjustment secured. This adjustment should be carefully made, using a thin tissue paper of not over .003-inch thickness to check the fit between the throttle valve and the barrel. Move the float up and down to see that its movement is entirely free. Check the movement and condition of the mixture control valve. Thoroughly examine and check all the parts for wear or irregularities. Replace by new parts where necessary. Reassemble the lower half of the carburetor, using the instruction sheets furnished for the individual model.

Before installing the needle valve and seat, they should be checked for leakage by holding the needle point upward with the seat in place and filling the small space above the needle with gasoline. If leakage is evident, the needle valve should be lapped in with crocus powder. If a good seat is not obtained after lapping, a new needle valve and seat should be used. It has been found in service that brass plugs screwed into aluminum have a tendency to stick or hold if they have been in place for a long period of time. This trouble can be obviated by the application of a thin coating of graphite oil paste to the threads of the plugs or other parts before screwing into place. The paste is made by mixing powdered graphite and

castor oil together to give a compound with a consistency such that it will just flow. Only a thin coating should be applied and care must be taken to avoid forcing any of the mixture into the carburetor.

Checking Float Level.—After assembling the lower half of the carburetor, the float level should be checked and this is done most easily by holding the lower half rigidly in a level position with a vise or other means, admitting fuel under a four or five foot gravity head through the regular inlet connection and measuring the distance from the level in the float chamber to the parting surface. This measurement can be made either by means of a scale between the surfaces directly or by use of a glass tube or gauge glass connected to the float chamber. The use of a gauge glass is the preferred method, particularly where a number of carburetors are to be checked. In case of an incorrect fuel level, adjustment should be made by changing the thickness of gaskets under the float needle seat. If the float needle is between the float and the pivot, increasing the gasket thickness will raise the level and vice-versa. If, however, the pivot is between the float and the needle, the action is reversed and increasing the gasket thickness will lower the level. The amount of change varies with different lengths of float brackets but a change of $\frac{1}{64}$ inch in gasket thickness will change the level approximately $\frac{5}{64}$ inch. Great care should be exercised in removing and replacing the needle seat, both to see that the gaskets are not misplaced or lost and that the needle valve seat is screwed tightly in place. A broad screw driver is required for this operation. After the correct level has been obtained, the carburetor should be allowed to stand with the fuel line connected for twenty or thirty minutes and the fuel level measured again. This is a check for a "creeping level." If the level has risen, drain the carburetor, remove the needle valve and seat and follow the procedure given for a leaky valve. Reassemble the complete carburetor, taking care to see that all gaskets are in place and that all loose parts are "safetied."

Bench Inspection.—When a carburetor is being changed from one engine to another or in any case where the condition is known to be good and it is desired merely to make a simple inspection to obtain that factor of safety necessary in all aircraft work, the following procedure should be gone through: The carburetor halves should be separated by parting at the joining surface. All plugs, screws, etc., at the ends of fuel passages should be removed. All fuel passages should then be examined as given in detail above under "Complete Inspection and Overhaul." Check the setting against that marked on the setting plate. See that the throttle valves fit the barrels and that the float movement is free. Test the mixture control valve for proper assembly and movement. Examine all moving parts for wear or irregularities. Clean any dirty parts and then reassemble the carburetor, taking the necessary precautions to see that all gaskets are in place and all loose parts "safetied."

Construction of Metering Jets.—The construction of the Stromberg metering jet (Fig. 135 A), combines the advantage of extreme accuracy and the ability to withstand undamaged the injuries incidental to hard service. The metering orifice proper is of the so-called "thin-plate" type which has the smallest variation of discharge coefficient with temperature

and viscosity changes of any type known. On account of the short length of orifice, it can be made very accurate throughout this length, and the approaches to the orifice can be made smooth and uniform. The location of the metering orifice in the middle of the body protects it from all injuries from tools. Its form and size are such that in the design of a carburetor it can be worked into an accessible location. For removal or replacement

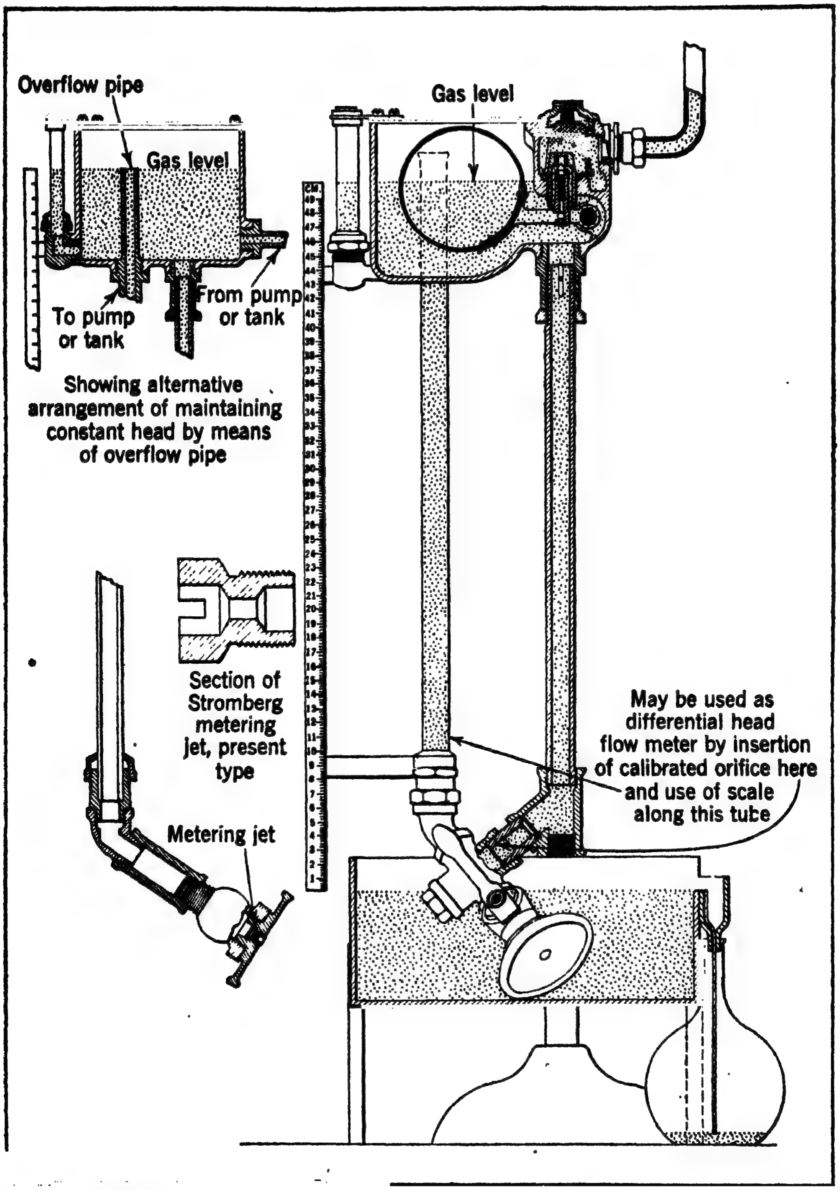


Fig. 135.—Typical Arrangement for Testing Metering Jet Capacity Under Condition of Submersion, has a Normal Carburetor Operation. Note Absence of Horizontal Passages where Air Bubbles Might Gather.

it requires no special tools, being screwed in and out of place with an ordinary screw driver. It is recommended that wire or other metal aids never be used in removing a metering jet from a carburetor as even soft wire will damage the metering orifice. A round stick of soft wood with a gradual taper and small diameter less than the diameter of the orifice will be found very good for this.

TABLE OF THREAD STANDARDS USED IN STROMBERG AIRCRAFT CARBURETORS

U. S. Standard Thread Form

No. 8-32	1/4"-20	3/8"-24	11/16"-24.
No. 10-24	1/4"-24	7/16"-24	3/4"-20
	1/4"-32	1/2"-24	7/8"-24
	5/16"-18	9/16"-24	1 1/2"-18
	5/16"-24	5/8"-24	

Calibration of Metering Jets.—The metering jets are numbered according to the Twist Drill and Wire Gauge Standard. They are first drilled undersize with a special drill which is so constructed as to leave as little burr as possible at the edges of the orifice and then carefully reamed with a reamer .0005-inch undersize. The calibration is made, however, on the basis of the flow capacity under standard conditions and not on the diameter of the metering orifice. The table that follows gives the standard flow data for these metering jets. These data are for a constant head of 50 centimeters and at normal room temperature. They apply with either aviation gasoline or water as a medium. These results were obtained with a flowmeter of the general construction illustrated in Fig. 135, and it is believed that the jets can be checked on any similar instrument. As shown, the jets are flowed completely submerged and it has been found that satisfactory results can only be obtained when the receiving basin is of sufficient size so that the surface of liquid in the basin is not disturbed by the jet flow. The design of a flow calibration meter should be such that it is impossible for air pockets or bubbles to form in any part.

Intake Manifold Design and Construction.—On four- and six-cylinder engines and in fact on all multiple-cylinder forms, it is important that the piping leading from the carburetor to the cylinders be made in such a way that the various cylinders will receive their full quota of gas and that each cylinder will receive its charge at about the same point in the cycle of operations. In order to make the passages direct the bends should be as few as possible, and when curves are necessary they should be of large radius because an abrupt corner will not only impede gas flow but will tend to promote condensation of the fuel. Every precaution should be taken with four- and six-cylinder engines to insure equitable gas distribution to the valve chambers if regular action of the powerplant is desired. If the gas pipe has many turns and angles it will be difficult to charge all cylinders properly. On some six-cylinder aviation engines, two carburetors are used because of trouble experienced with manifolds designed for one carburetor. Duplex carburetors are necessary to secure the best results from eight- and twelve-cylinder Vee engines, and in a twelve-cylinder the best results are obtained by using two duplex carbu-

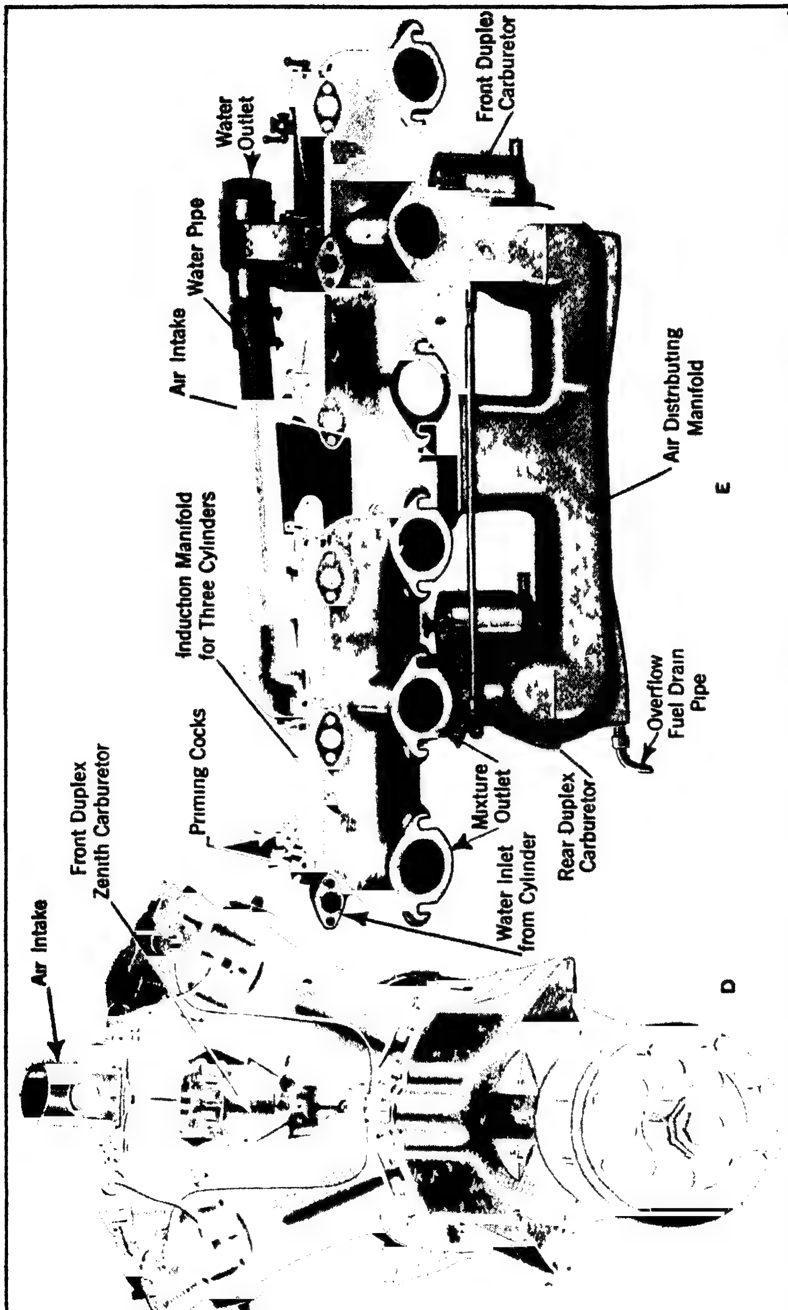


Fig. 136.—How Zenith Carburetor is Mounted on Liberty Engine Between Cylinder Banks Shown at D. E—Carburetor and Induction Manifold of Liberty Airplane Motor Removed from Cylinder.

retors installed as shown at Fig. 136. The carburetor location is inside the Vee, the manifold and carburetors removed from the Vee between the cylinders which they nearly fill is shown at Fig. 136 E. Carburetor location depends upon the engine design as shown at Fig. 137, which shows three types of Packard engines. In the conventional Vee form, the carburetor and manifold assembly is carried in the Vee. In the inverted Vee form, the manifold and carburetors are carried below the engine cylinders. In the 24-cylinder X engine, the methods used in each of the other types are used, one for the cylinders above the center line and the other for the inverted cylinders below it.

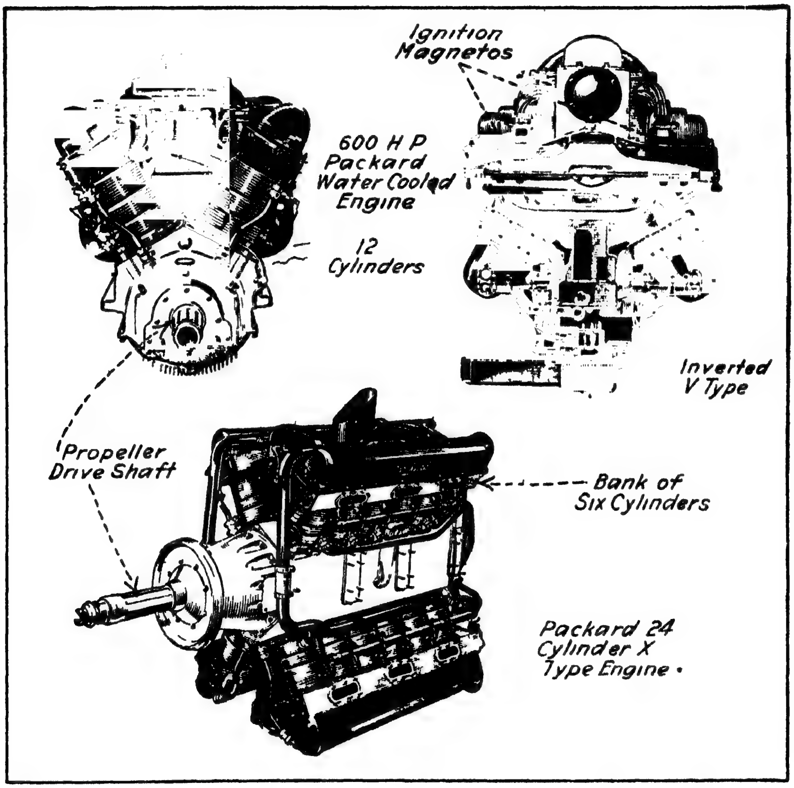


Fig. 137.—Packard Water-Cooled Aviation Engines Showing Method of Carburetor Mounting.

The problem of intake piping is simplified to some extent on block motors where the intake passage is cored in the cylinder casting and where but one short pipe is needed to join this passage to the carburetor. If the cylinders are cast in pairs a simple pipe of T or Y form can be used with success. When the engine is of a type using individual cylinder castings, especially in the six-cylinder powerplants, the proper application and installation of suitable piping is a difficult problem. In some of the twelve-cylinder Vee engines designed abroad the carburetors are carried outside the crankcase, as shown at Fig. 138 which shows the transverse

section of the Lorraine Vee engine. Duplex carburetors are carried each side of the crankcase, to which they are securely attached. Long mixture pipes extend to manifolds attached to the inner side of the cylinders, each manifold feeding three cylinders. In radial cylinder engines, as shown at Fig. 139, the problem is easily solved by coring a mixture distribution

STROMBERG METERING JETS SUBMERGED FLOW UNDER A CONSTANT HEAD OF 50 CENTIMETERS

With either Water or Aviation Gasoline at Temperatures between 55° and 75° F.

Jet Size No.	Diameter	Flow in Cubic Centimeters per Minute	Flow in U. S. Pints per Hour
60	.0400	126	16.0
59	.0410	134	17.0
58	.0420	139	17.6
57	.0430	147	18.6
56	.0465	170	21.5
55	.0520	215	27.2
54	.0550	239	30.3
53	.0595	290	36.8
52	.0635	320	40.6
51	.0670	360	45.7
50	.0700	400	50.8
49	.0730	432	54.8
48	.0760	474	60.1
47	.0785	510	64.6
46	.0810	545	69.1
45	.0820	561	71.0
44½ (2.15 M-M)	.0847	605	76.8
44	.0860	628	79.7
43	.0890	678	86.0
42½ (2.30 M-M)	.0906	705	89.5
42	.0935	755	95.7
41	.0960	800	101.5
40	.0980	836	106.0
39	.0995	866	110.0
38	.1015	912	115.8
37	.1040	958	121.7
36	.1065	1,010	128.0
35	.1100	1,088	138.0
34	.1110	1,110	140.8
33	.1130	1,151	146.0
32	.1160	1,225	155.5
31	.1200	1,322	167.8
30½ (⅛-inch)	.1250	1,450	184.0
30	.1285	1,550	196.8
29½ (3.40 M-M)	.1340	1,710	216.8
29	.1360	1,770	224.4
28	.1405	1,900	240.5
27	.1440	2,015	255.0

passage in the crankcase to which the carburetor is bolted and having pipes extend from this distribution chamber to each cylinder. The reader is referred to the various engine designs outlined to ascertain how the inlet piping has been arranged on representative aviation engines.

Intake piping is constructed in two ways, the most common method being to cast the manifold of aluminum. The other method, which is more

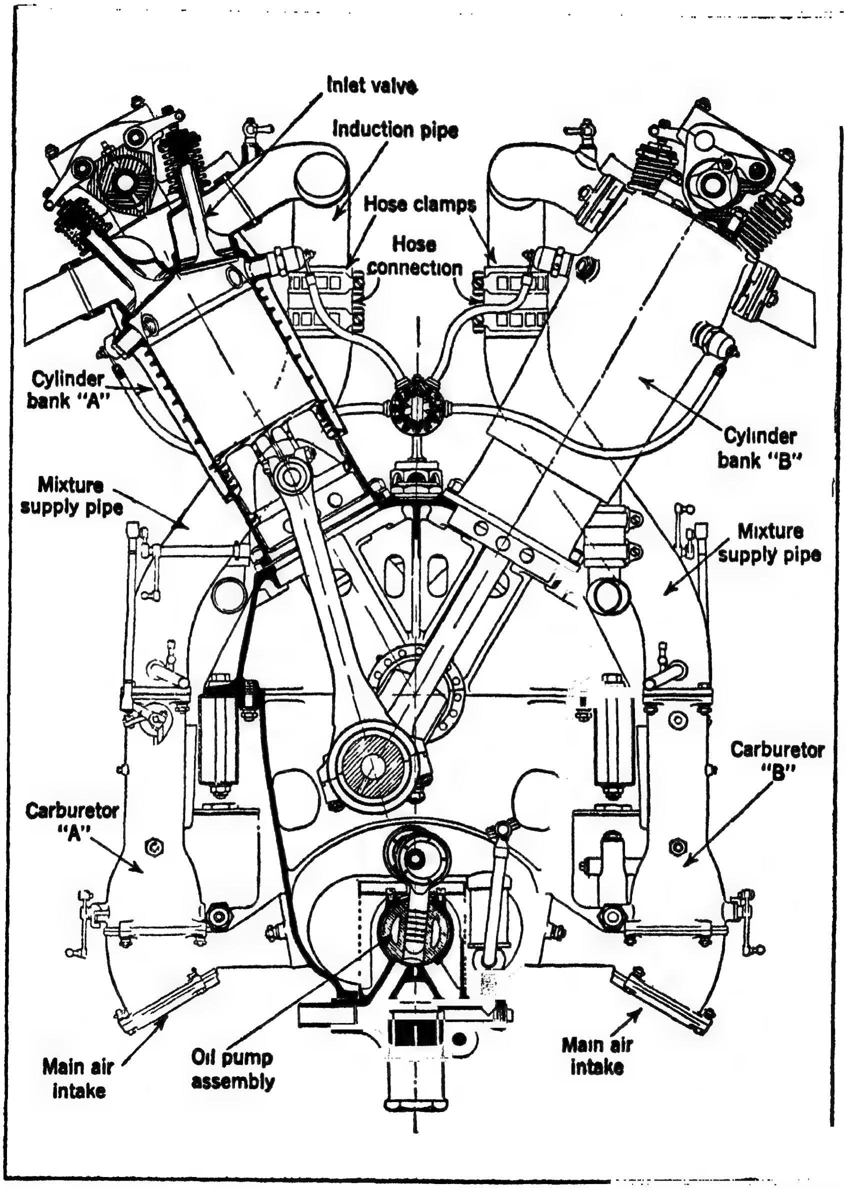


Fig. 138.—Transverse Sectional View of the Lorraine "Vee" Engine Showing Use of Carburetors Mounted Outside of the Engine Base, Supplying Mixture Through Long Induction Pipes.

costly, is to use a built-up construction of very thin wall copper, brass or Dural tubing with cast metal elbows and Y pieces. One of the disadvantages advanced against the cast manifold is that blowholes may exist which producé imperfect castings and which will cause mixture troubles because the entering gas from the carburetor, which may be of proper proportions, is diluted by the excess air which may leak in through the porous casting. Another factor of some moment is that the roughness of the cast walls has a certain amount of friction which tends to reduce

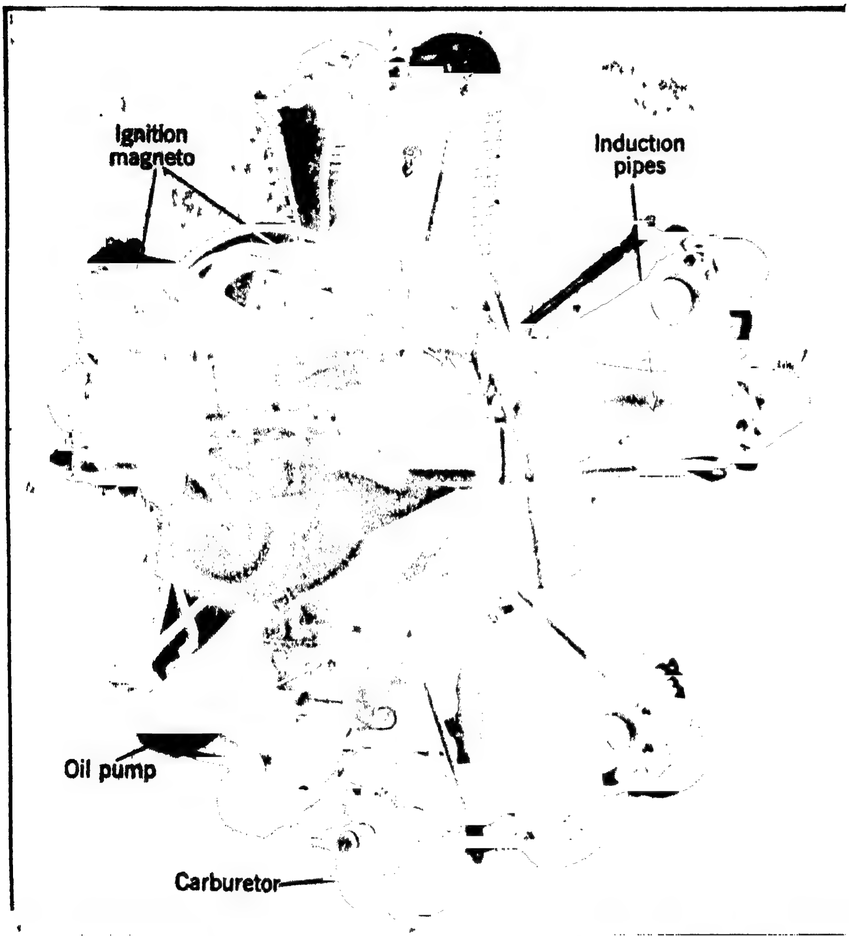


Fig. 139.—Anti-Propeller End of LeBlond Five-Cylinder Static Radial Engine Showing Installation of Ignition Magnetos and Zenith Aircraft Carburetor.

the velocity of the gases, and when projecting pieces are present, such as core wire or other points of metal, these tend to collect the drops of liquid fuel and thus promote condensation. The advantage of the built-up construction is that the walls of the tubing are very smooth, and as the castings are small it is not difficult to clean them out thoroughly before they are incorporated in the manifold. The tubing and castings are joined together by hard soldering, brazing or autogenous welding.

Compensating for Various Atmospheric Temperatures.—The low-grade gasoline used at the present time makes it necessary to use vaporizers that are more susceptible to atmospheric variations than when higher grade and more volatile liquids are vaporized. Sudden temperature changes, sometimes being as much as 40 degrees rise or fall in twelve hours, affect the mixture proportions to some extent, and not only changes in temperature but variations in altitude also have a bearing on mixture proportions by affecting both gasoline and air. As the temperature falls the specific grav-

TABLE OF TWIST DRILL AND STEEL WIRE GAUGE SIZES AND AREAS, TOGETHER WITH NEAREST EQUIVALENT METRIC SIZES

Drill No.	Dia., In.	Area, Sq. In.	Nearest Metric Size, MM.	Dia., In.	Area, Sq. In.	Drill No.	Dia., In.	Area, Sq. In	Nearest Metric Size, MM.	Dia., In.	Area, Sq. In.
1	.2280	.04082	5.8	.2298	.04093	41	.0960	.00723	2 45	.0965	.00731
2	.2210	.03835	5.6	.2205	.03819	42	.0935	.00686	2.35	.0925	.00672
3	.2130	.03563	5.4	.2127	.03553	43	.0890	.00622	2.25	.0886	.00617
4	.2090	.03430	5.3	.2087	.03421	44	.0860	.00580	2.20	.0866	.00589
5	.2055	.03316	5.2	.2047	.03291	45	.0820	.00528	2.10	.0827	.00537
6	.2040	.03268	5.2	.2047	.03291	46	.0810	.00515	2.05	.0807	.00512
7	.2010	.03183	5.1	.2008	.03167	47	.0785	.00483	2.00	.0787	.00486
8	.1990	.03110	5.1	.2008	.03167	48	.0760	.00453	1.95	.0768	.00463
9	.1960	.03017	5.0	.1967	.03039	49	.0730	.00418	1.85	.0728	.00416
10	.1935	.02932	4.9	.1929	.02918	50	.0700	.00384	1.80	.0709	.00395
11	.1910	.02865	4.9	.1929	.02918	51	.0670	.00352	1.70	.0669	.00352
12	.1890	.02805	4.8	.1890	.02805	52	.0635	.00316	1.60	.0630	.00312
13	.1850	.02688	4.7	.1850	.02688	53	.0595	.00278	1.50	.0591	.00274
14	.1820	.02601	4.6	.1811	.02576	54	.0550	.00237	1.40	.0551	.00238
15	.1800	.02544	4.6	.1811	.02576	55	.0520	.00212	1.30	.0512	.00206
16	.1770	.02460	4.5	.1772	.02456	56	.0465	.00169	1.20	.0472	.00175
17	.1730	.02350	4.4	.1732	.02356	57	.0430	.00145	1.10	.0433	.00147
18	.1695	.02256	4.3	.1693	.02251	58	.0420	.00138	1.05	.0413	.00134
19	.1660	.02164	4.2	.1654	.02149	59	.0410	.00132	1.05	.0413	.00134
20	.1610	.02035	4.1	.1614	.02046	60	.0400	.00125	1.00	.0394	.00122
21	.1590	.01985	4.0	.1575	.01948	61	.0390	.00119	1.00	.0394	.00122
22	.1570	.01862	4.0	.1575	.01948	62	.0380	.00113	.95	.0374	.00110
23	.1540	.01842	3.9	.1535	.01851	63	.0370	.00107	.95	.0374	.00110
24	.1520	.01814	3.9	.1535	.01851	64	.0360	.00101	.90	.0354	.00098
25	.1495	.01755	3.8	.1496	.01758	65	.0350	.00096	.90	.0354	.00098
26	.1470	.01697	3.7	.1457	.01667	66	.0330	.00085	.85	.0335	.00088
27	.1440	.01628	3.7	.1457	.01667	67	.0320	.00080	.80	.0315	.00078
28	.1405	.01550	3.6	.1417	.01577	68	.0310	.00075	.80	.0315	.00078
29	.1360	.01452	3.5	.1378	.01491	69	.0293	.00067	75	.0295	.00068
30	.1285	.01296	3.3	.1299	.01325	70	.0280	.00061	.70	.0276	.00060
31	.1200	.01130	3.0	.1181	.01094	71	.0260	.00053	.65	.0256	.00051
32	.1160	.01056	2.9	.1142	.01024	72	.0250	.00049	.65	.0256	.00051
33	.1130	.01002	2.9	.1142	.01024	73	.0240	.00045	.60	.0236	.00044
34	.1110	.00967	2.8	.1102	.00954	74	.0225	.00039	.55	.0217	.00037
35	.1100	.00950	2.8	.1102	.00954	75	.0210	.00034	.55	.0217	.00037
36	.1065	.00890	2.7	.1063	.00887	76	.0200	.00031	.50	.0197	.00030
37	.1040	.00849	2.6	.1024	.00824	77	.0180	.00025	.45	.0177	.00025
38	.1015	.00809	2.6	.1024	.00824	78	.0160	.00020	.40	.0157	.00019
39	.0995	.00777	2.5	.0984	.00760	79	.0145	.00016	.35	.0138	.00015
40	.0980	.00754	2.5	.0984	.00760	80	.0135	.00014	.35	.0138	.00015

ity of the gasoline increases and it becomes heavier, this producing difficulty in vaporizing. The tendency of very cold air is to condense gasoline instead of vaporizing it and therefore it is necessary to supply heated air to some carburetors to obtain proper mixtures during cold weather. In order that the gas mixtures will ignite properly the fuel must be vaporized and thoroughly mixed with the entering air either by heat or high velocity of the gases. The application of air stoves to the Curtiss OX2 motor is clearly shown at Fig. 113. It will be seen that flexible metal pipes are used to convey the heated air to the air intakes of the duplex Zenith carburetor mixing chamber.

The Wright J 5 Air Stove.—The problem of carburetion on the Wright Whirlwind engine is intimately connected with the quality of fuel used. With the best grades of aviation gasoline no difficulty is experienced with the smoothness or acceleration of the Whirlwind engine at any speed and it is possible to run on extremely lean mixtures. With the poorer grades of aviation gasoline a hesitation in acceleration is noticeable and rough running is obtained usually between 1,400 and 1,600 r.p.m. on a propeller which turns 1,800 r.p.m. at full throttle. In cold weather this condition becomes worse and can be only partially prevented by the use of rich mixtures. The presence of ice in the carburetor is another source of annoyance and even danger to the pilot. When the humidity is high, even in hot weather, the evaporation of gasoline drops the temperature of the mixture below the freezing point and the moisture contained in the air condenses on the manifold walls and freezes. This ice gradually builds up and chokes off the manifold passage, eventually stopping the engine.

To eliminate the operating difficulties mentioned above a carburetor air heater has been developed. This heater consists of an aluminum casting, as shown at Fig. 140, which bolts directly to the carburetor flange. Incorporated in the casting is a passage for exhaust gas which runs horizontally at right angles to the engine crankshaft. This passage is finned on both the inside and outside to obtain the maximum heat transfer from the exhaust gas to the intake air. Above the heating element a valve is fitted which when open will shut off the air passing the heating element and open a port admitting unheated air to the carburetor. The exhaust from cylinders Nos. 5 and 6 is brought together in a double elbow with a pipe to discharge the exhaust gas into a manifold or into the slipstream as desired. The exhaust connections to the heater are made with elbows of steel tubing welded to the exhaust flanges. The elbows are so designed that a straight piece of flexible metal tubing can be clamped between the elbow on the carburetor air heater and the elbow on the exhaust port. The flexible tubing is secured on the ends with hose clamps which bind it tightly to the steel elbow.

The bypass valve in the heater is designed to close at part throttle so that the maximum amount of heat is utilized up to the full throttle position. At full throttle the valve is opened and the carburetor operates with no heat; thus the maximum power of the engine is unaffected by the use of the heater. The effect of heat at part throttle is to reduce the volumetric efficiency and thus slightly reduce the power for any given throttle position. This has exactly the same effect as a slightly smaller opening of

the throttle valve except that the heat supplied vaporizes the incoming fuel and causes better distribution to all cylinders, thus the effect of the heater is to slightly improve the fuel consumption due to better distribution. The power of the engine is not affected as the throttle can be set to a slightly greater opening to compensate for the loss of volumetric efficiency. The weight of the heater and piping is seven pounds. This includes the elbows and flexible tubing, some of which would be duplicated if exhaust

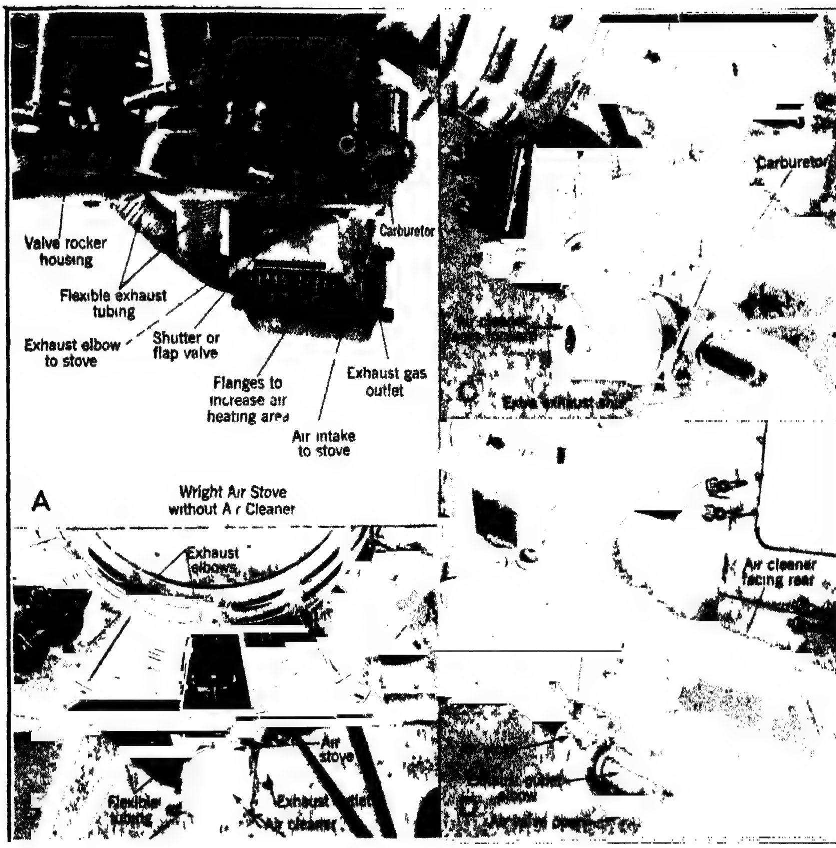


Fig. 140.—How Wright Carburetor Air Intake Heater is Installed by Using Two Exhaust Elbows Shown at A. Side View of Air Intake Heater Showing Cast Aluminum Fins Inside and Outside the Passage for the Hot Exhaust Gas.

manifolds were fitted Considering the greatly improved performance and reliability of the Whirlwind engine due to the use of this heater, the added weight is considered insignificant. This heater is now furnished as standard equipment on all J 5 Whirlwind engines but cannot be adapted to the J 4B or earlier types of Whirlwinds due to a change in carburetor model.

Installation of P and W Wasp Mixture Heater.—The need of air stoves is now generally recognized by nearly all builders of radial engines as a useful adjunct. The Pratt and Whitney Aircraft Company state in their instruction manual that satisfactory "Wasp" engine performance in cold weather requires a mixture heater, and this device is furnished with

each Wasp engine. The heater is located between the carburetor and the rear section of the crankcase, and consists of an exhaust heated section of the intake system. Any fuel which may leave the carburetor in a liquid state is vaporized by the heater before entering the supercharger. It has been found that the application of heat is much more effective at this point than before the air enters the carburetor. Cold weather operation without the heater will cause rough running due to poor distribution. Raw fuel is fed to certain cylinders, causing them to run rich, while other cylinders will run too lean. Continued operation under these conditions will result in high fuel consumption and piston failures. The use of the hot spot mixture heater shortens the warming-up period, makes the engine

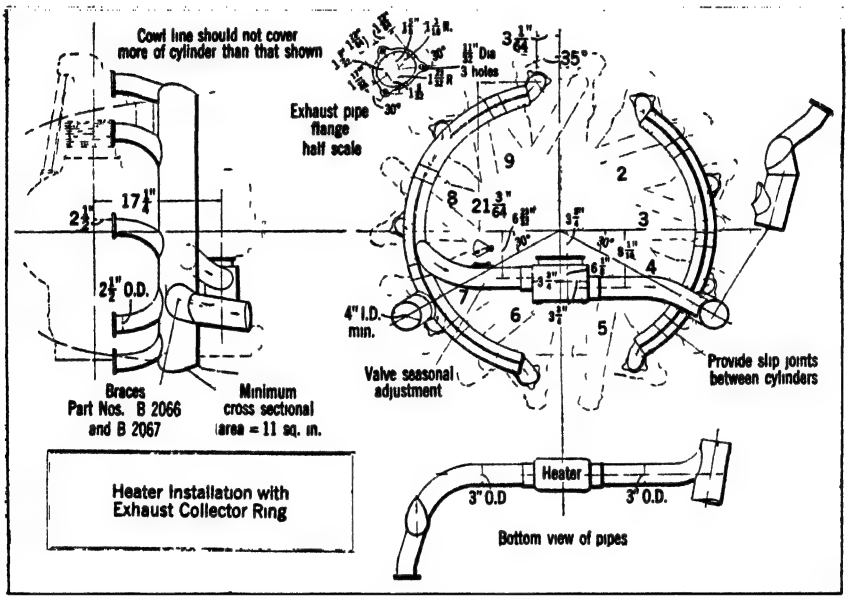


Fig. 141A.—Heater Installation with Exhaust Collecting Ring of Pratt & Whitney "Wasp" Engines.

run smoothly, and lowers the fuel consumption considerably. Another important advantage is the elimination of cutting out after a long glide. Sufficient heat is stored in the heater to vaporize the fuel immediately when the throttle is opened. The operation of the heater is automatic and requires no attention from the pilot. A seasonal control valve may be fitted, to reduce the heat supply in warm weather, but it is not necessary to have this controlled from the pilot's seat.

The heater is provided with an asbestos inner covering and an aluminum outside jacket. Braces are furnished to steady the carburetor. The piping to conduct exhaust gases into the heater depends altogether on the individual installation, and is supplied by the airplane builder. Drawings of typical installations are shown at Figs. 141 A and 141 B. With either individual exhaust stacks or a complete collector ring, the piping should be arranged so that the heater does not form the sole outlet for any exhaust port, and if a control valve is provided it may be put in the pipe which feeds

the heater. This will allow the exhaust to be shut off from the heater, causing it to take the alternative means of escape. More detailed information regarding the use and installation of the hot spot heater may be had by addressing the Engineering Service Department, Pratt and Whitney Aircraft Co.

Utility of Fuel Strainers.—Many carburetors include a filtering screen at the point where the liquid enters the float chamber in order to keep dirt or any other foreign matter which may be present in the fuel from entering the float chamber. This is now general practice at this time and the majority of vaporizers do include a filter in their construction. It is very desirable that the dirt should be kept out of the carburetor because it may get under the float control fuel valve and cause flooding

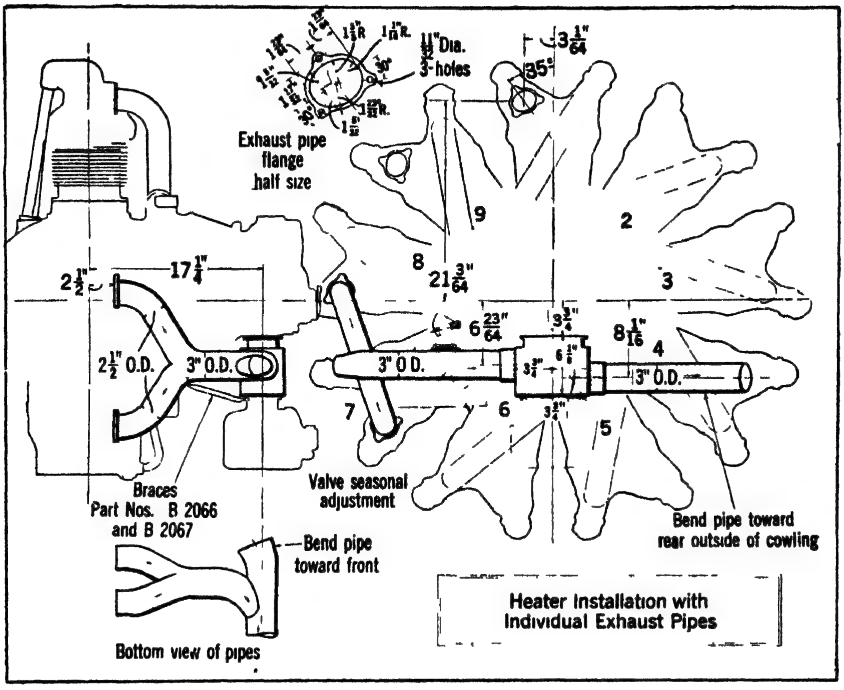


Fig. 141B.—Carburetor Air Heater Installation on "Wasp" Engine with Individual Exhaust Pipes.

by keeping it raised from its seat. If it finds its way into the spray nozzle it may block the opening so that no gasoline will issue or may so constrict the passage that only very small quantities of fuel will be supplied the mixture. Where the carburetor itself is not provided with a filtering screen a simple filter is usually installed in the pipe line between the gasoline tank and the float chamber and these strainers have been made in wide variety. Some simple forms of filters and separators are shown at Fig. 142. That at A consists of a simple brass casting having a readily detachable gauze screen and a settling chamber of sufficient capacity to allow the foreign matter to settle to the bottom, from which it is drained out by a pet cock. Any water or dirt in the gasoline will settle to the

bottom of the chamber, and as all fuel delivered to the carburetor must pass through the wire gauze screen it is not likely to contain impurities when it reaches the float chamber. The heavier particles, such as scale from the tank or dirt and even water, all of which have greater weight than the gasoline, will sink to the bottom of the chamber, whereas light particles, such as lint, will be prevented from flowing into the carburetor by the filtering screen. The filtering device shown at B is a larger appliance than that shown at A, and should be more efficient as a separator because the gasoline is forced to pass through three filtering screens before it reaches the carburetor. The gasoline enters the device shown at

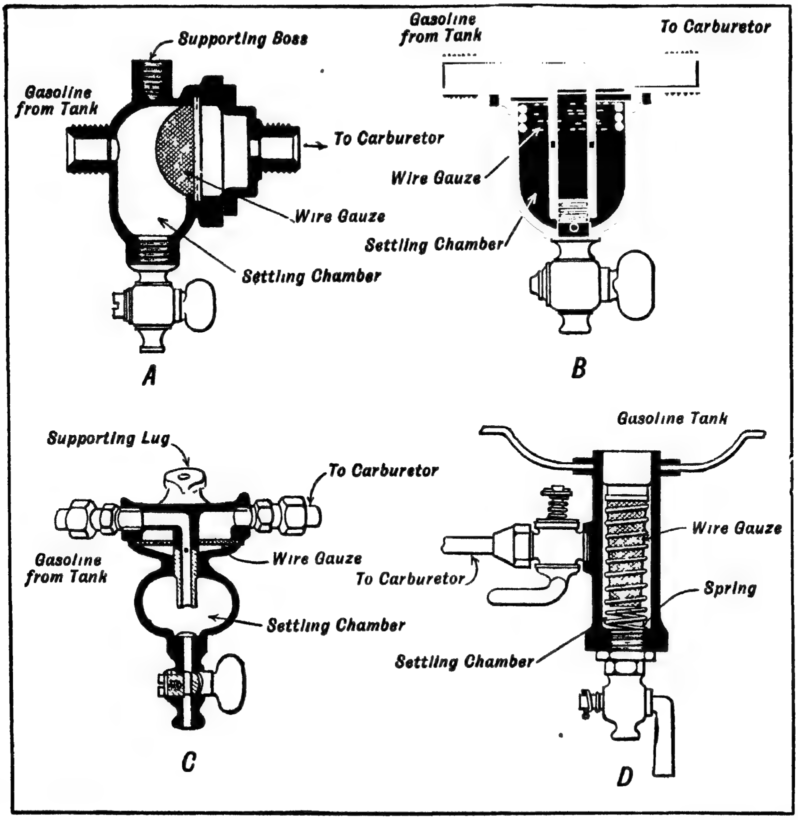


Fig. 142.—Types of Strainers Interposed Between Vaporizer and Gasoline Tank to Prevent Water or Dirt Passing into Carburetor.

C through a bent pipe which leads directly to the settling chamber and from thence through a wire gauze screen to the upper compartment which leads to the carburetor. The device shown at D is a combination strainer, drain, and sediment cup. The filtering screen is held in place by a spring and both are removed by taking out a plug at the bottom of the device. The shut-off valve at the top of the device is interposed between the sediment cup and the carburetor. This separating device is incorporated with

the gasoline tank and forms an integral part of the gasoline supply system. The other types shown are designed to be interposed between the gasoline tank and the carburetor at any point in the pipe line where they may be conveniently placed.

QUESTIONS FOR REVIEW

1. What are the rules for determining Stromberg carburetor settings?
2. How is Venturi tube size determined?
3. How is main metering jet size found?
4. When is it necessary to change accelerating well bore?
5. Give rules for successful carburetor installation.
6. What is normal engine starting procedure?
7. Outline routine carburetor inspection in airplane.
8. How is metering jet capacity bench tested?
9. Name some precautions to be observed in inlet manifold construction.
10. What is the value of an air stove; of a fuel filtering device?

CHAPTER XII

AIRCRAFT ENGINE SUPERCHARGERS—DIESEL ENGINES

Why High Altitude Affects Power Output—Airplane Engine Superchargers—Roots Type Compressor—Farman Supercharged Engine—Pressure or Suction Supercharging—Air Corps Supercharger Development—Efficiency of Centrifugal Superchargers—Blowers for Charge Distribution—Superchargers Aid Scavenging—Practical Value of Superchargers—Automotive Diesel Engines—Peugeot-Junkers Type Diesel Engine—Fuel Injection a Problem—Sperry Oil Engine for Aircraft.

Any internal-combustion engine will show less power at high altitudes than it will deliver at sea level, and this has caused a great deal of questioning. There is a good reason for this and it is a physical impossibility for the engine to do otherwise. The difference is due to the lower atmospheric pressure the higher up we get. That is, at sea level the atmosphere has a pressure of 14.7 pounds per square inch; at 5,000 feet above sea level the pressure is approximately 12.13 pounds per square inch, and at 10,000 feet it is ten pounds per square inch. From this it will be seen that the final pressure attained after the piston has driven the gas into compressed condition ready for firing is lower as the atmospheric pressure drops. This means that there is not so much power in the compressed charge of gas the higher up you get above sea level.

Why High Altitude Affects Power Output.—For example, suppose the compression ratio to be 4½ to 1; in other words, suppose the air space above the piston to have 4½ times the volume when the piston is at the bottom of its stroke that it has when the piston is at the top of the stroke. That is a common compression ratio for an average motor, and is chosen because it is considered to be the best for maximum horsepower and in order that the compression pressure will not be so high as to cause pre-ignition. Knowing the compression ratio, we can determine the final pressure immediately before ignition by substituting in the standard formula:

$$P^1 = P\left(\frac{V}{V^1}\right)^{1.3}$$

in which P is the atmospheric pressure; P^1 is the final pressure, and $\frac{V}{V^1}$ is the compression ratio, therefore $P^1 = 14.7\ (4.5)^{1.3} = 104$ pounds per square inch, absolute.

That is, 104 pounds per square inch is the most efficient final compression pressure to have for this engine at sea level, since it comes directly from the compression ratio.

Now supposing we consider that the altitude is 7,000 feet above sea level. At this height the atmospheric pressure is 11.25 pounds per square inch, approximately. In this case we can again substitute in the formula,

using the new atmospheric pressure figure. The equation becomes:

$P^1 = 11.25\ (4.5)^{1.3} = 79.4$ pounds per square inch, absolute.

Therefore we now have a final compression pressure of only 79.4 pounds per square inch, which is considerably below the pressure we have just found to be the most efficient for the motor. The resulting power drop is evident.

It should be borne in mind that these final compression pressures are absolute pressures—that is, they include the atmospheric pressure. In the first case, to get the pressure above atmospheric you would subtract 14.7 and in the latter 11.25 would have to be deducted. In other words, where the sea level compression is 89.3 pounds per square inch above the atmosphere, the same motor will have only a compression pressure of 68.15 pounds per square inch above the atmosphere at 7,000 feet elevation.

From the above it is evident that in order to bring the final compression pressure up to the efficient figure we have determined, a different compression ratio would have to be used. That is, the final volume would have to be less, and as it is difficult to vary this to meet the conditions of altitude, the loss of power cannot be helped except by the replacing of the standard pistons with some that are longer above the wristpin so as to reduce the space above the pistons when on top center. Then if the ratio is thereby raised to some such figures as five to one, the engine will again have its proper final pressure, but it will still not have as much power as it would have at sea level, since the horsepower varies directly with the atmospheric pressure, final compression being kept constant. That is, at 7,000 feet the horsepower of an engine that had 40 horsepower at sea level would be equal to

$$\frac{11.25}{14.7} = 30.6 \text{ horsepower.}$$

If the original compression ratio of 4.5 were retained, the drop in horsepower would be even greater than this. These computations and remarks will make it clear that the designer who contemplates building an airplane for high altitude use should see to it that it is of sufficient power to compensate for the drop that is inevitable when it is up in the air. This is often illustrated in stationary gas-engine installations. An engine that had a sea-level rating amply sufficient for the work required, might not be powerful enough when brought up several thousand feet. When one considers that airplanes that are not supercharged attain heights of over 20,000 feet, it will be evident that an ample margin of engine power at sea level is necessary to attain such a ceiling without supercharging.

Considerable data dealing with the altitude performance of the internal-combustion engine has been published, and it is not planned to discuss this phase of the problem in this treatise to any extent. It is enough to call attention to the fact that, as the power developed in the cylinders of the internal-combustion engine is directly proportional, other conditions being constant, to the weight of charge burned in a unit of time, it naturally follows that the power developed at altitude, for a constant engine speed, will decrease approximately in direct proportion with the decreased air-

density; the brake, or effective, power decreasing somewhat more rapidly due to the fact that certain of the friction losses remain constant. Thus, an engine will develop somewhat less than one-half its sea level power under altitude conditions of one-half air-density, or at an altitude about 20,000 feet because the weight of mixture taken into the cylinders is only one-half as great.

Airplane Engine Superchargers.—An examination of some of the present day engines and a review of the descriptive matter preceding would lead one to the opinion that, in the light of current knowledge of materials, a further important reduction in the specific weight of conventional four-stroke-cycle engines is hardly to be expected without sacrificing durability unless there be some radical change in design not at present contemplated. Therefore, with due regard for continued improvement along these lines and for the possible development of engine types that are radically different from the present accepted standards, it would seem that one of the most logical fields for improvement of the aircraft engine would be the bettering of its altitude performance. Supercharging means forcing into the engine cylinders a quantity of charge greater than can be drawn in by the mere displacement of the pistons under similar conditions of speed and atmospheric density. It is a very old idea which probably was suggested and tried out even before application to automobiles made it desirable to get the greatest possible power from internal-combustion cylinders of given dimensions.

During the war the idea of compensating for the natural decrease in power output of aircraft engines with increase in altitude suggested itself, and a great deal of research work and experimentation was done by all the leading belligerents. At an altitude of about 20,000 feet an ordinary aircraft engine develops only about one-half as much power as at ground level. This decrease in engine power with increase in altitude limits the maximum height that may be attained by the plane, and as in aerial encounters the machine which has the highest ceiling has a great advantage over its adversary, the importance of supercharging is obvious. Air can be forced into the cylinders only by means of a pump as shown at Fig. 143 and of the different types of pump known—plunger, rotary piston and centrifugal—only the latter two have been used on four-cycle aircraft engines so far. Two methods of driving have been developed. Rateau in France and Moss in this country drive the centrifugal pump or blower by means of a turbine operated by the exhaust gases from the engine, while others, use a direct mechanical drive from the engine. The reader should note an exception in the Attendu oil engine previously described where the air pump is of the piston type driven by an auxiliary shaft.

Both common methods of supercharger drive involve rather serious difficulties. With the exhaust turbine drive the weakening of the impeller blades by the high temperature to which they are exposed, combined with the high stresses to which they are subjected by the centrifugal force, calls for the most careful selection of material and the most rational proportioning of the blades or vanes. With a turbine drive there is also a back pressure on the exhaust, which has a tendency to reduce the engine output, and unless the apparatus is properly designed the net output may be decreased

instead of being increased. The reduced air-density at great altitudes offers reduced resistance to high airplane speeds; hence the same power that will drive a plane at a speed of 120 m.p.h. at sea level will drive it much faster at 20,000 feet, and still faster at 30,000 feet altitude, *and with approximately the same consumption of fuel per horsepower hour,* providing the area of the aerofoils and pitch of propeller are such as needed in the thinner air.

Superchargers usually take the form of a mechanical blower or pump and, of course, require a driving gear of some kind. The types of blowers or compressors used to date include the reciprocating, Roots displacement

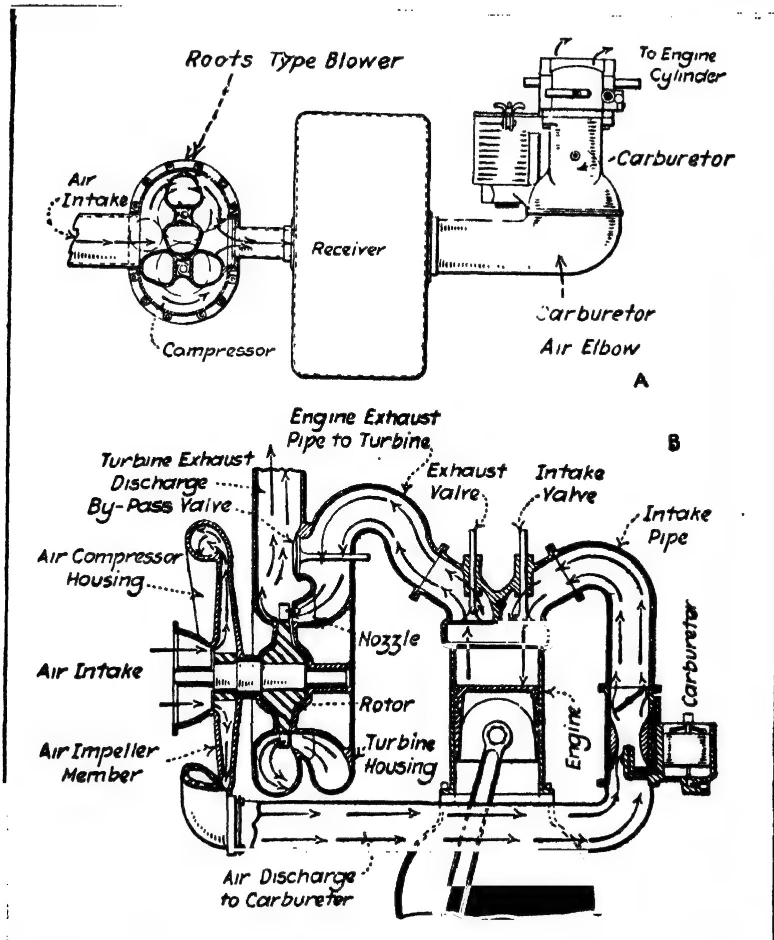


Fig. 143.—Two Types of Superchargers for Airplane Engines. A—Arrangement of Roots Blower. B—Application of Turbine Driven Centrifugal Impeller Air Blower. The Turbine Utilizes the Pressure of Exhaust Gases for Power. Note Location of Carburetor Relative to Blowers.

and centrifugal types. When the Roots type is employed, as shown at Fig. 143 A some form of receiver is necessary to equalize the pulsating nature of the discharge. This form, when driven by positive gearing,

may be timed so its greatest pressure may be coincidental with the induction stroke of the cylinder. The reciprocating type of piston pump is only suited for slow speed Diesel engines as it is much too heavy for airplane service except in the special Attendu application.

Each type has its proponents, and it seems that each type has its useful sphere. No attempt will be made to define their limits or merits, as that has caused endless discussion. In general, it is believed that the impeller type is suitable for low-altitude work but that it is totally unsuitable for

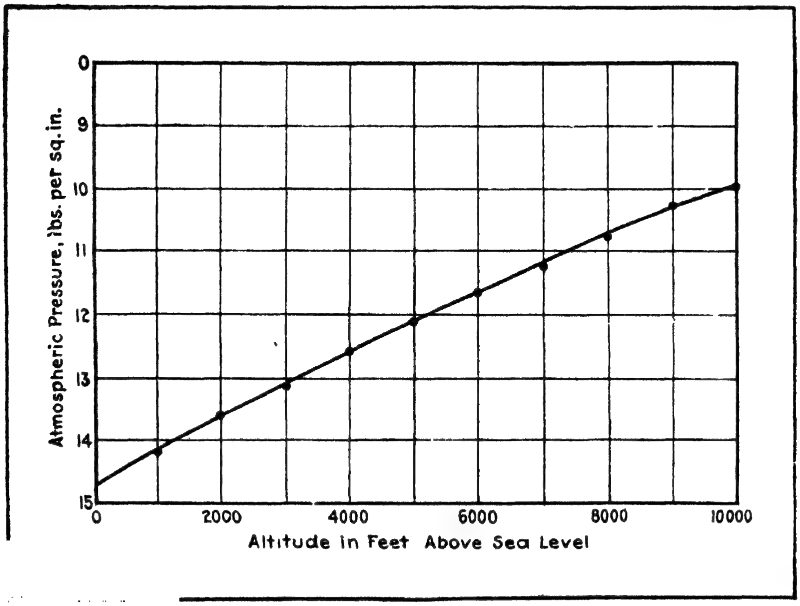


Fig. 143C.—Chart Showing Diminution of Air Pressure as Altitude Increases.

high-altitude work unless fitted with some form of variable-speed drive which, of course, adds weight and complication and detracts greatly from its attractiveness. For high-altitude supercharging, an impeller of reasonable size needs to be driven at extremely high speeds. The precessional forces set up in the fan due to rapid maneuvers of fast airplanes would be great. The heating of the charge results in a large loss in volumetric efficiency in the engine and requires an intercooler one-fourth the size of the radiating surface of the powerplant. The greatest heating and power absorption take place at sea level, where the supercharger cannot be used, and the compressed charge must be re-expanded by some wire-drawing at the throttle at all altitudes below the critical altitude to avoid excessive compression-pressures.

Roots Type Compressor.—The Roots type of compressor was a commercial product in 1859, when it was manufactured by the P. H. & P. M. Roots Co. of Connersville, Ind., from which company it probably received its name. The Roots blower seems well suited for supercharging and, in the form developed by the National Advisory Committee for Aeronautics,

has been highly satisfactory. This is illustrated in Fig. 144. Much laboratory work and experimental flight testing have been done and, at present, a squadron of new airplanes is being equipped with this type of supercharger. These superchargers have cast aluminum-alloy side-casings, heavily ribbed to prevent distortion under load, to permit the use of small rotor-clearances. The rotors are hollow magnesium castings, internally ribbed, and have splined steel-hubs on the shafts. The rotors are carried on ball-bearings, primarily to simplify the lubrication problem.

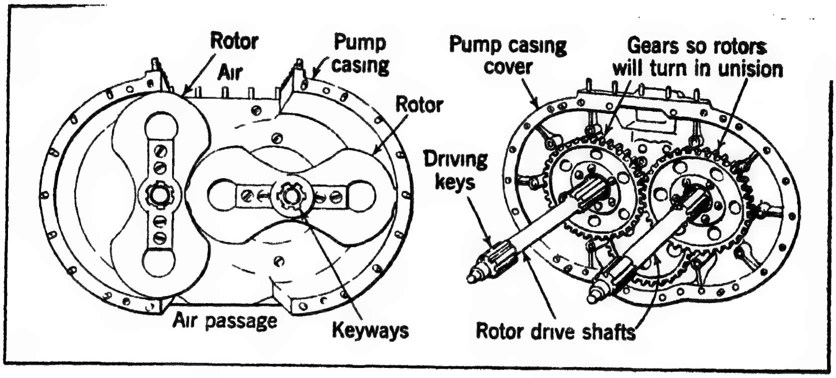


Fig. 144.—Interior of Roots Type Supercharger. The Driving Gears and Shaft of the Rotor are Shown at the Right. Note the Mesh of the Rotor and the Contour which Differs from that of the Customary Commercial Roots Blower, as Outlined at the Left of the Illustration.

The blower is driven through a flexible coupling at one and one-half to three times engine-speed, depending upon the critical altitude desired. Intake air is drawn in through a duct from outside the fuselage. The discharge is led directly to the carburetor through another duct, which has fitted in it at any convenient point a large bypass valve, the effective area of which is equal to the cross-sectional area of the duct. The valve is connected either to an extension of the throttle or to an independent lever, and constitutes the complete control of the supercharger. At sea level, or whenever no supercharging is desired, the valve is held wide-open and the carburetor can get air at atmospheric pressure corresponding to the altitude. The supercharger is then running in the "no-load" condition and is absorbing one or two horsepower to overcome friction and the "fanning effect." As the airplane ascends into air of less density, the valve is regulated to provide the desired degree of supercharging pressure at the carburetor. At any altitude below the critical altitude with the centrifugal compressor, the work done on the air is greater than that done by the Roots blower type; hence, the power absorbed and the temperature rise of the charge are greater. With a turbine drive, the hot-exhaust collector is very objectionable.

The efficiency of the two types as compressors is approximately the same. For rotary distribution as with radial cylinder engines the centrifugal blower is ideal. It puts back into the mixture the heat absorbed in vaporization of the fuel and, as it is gear-driven, it prevents any increase

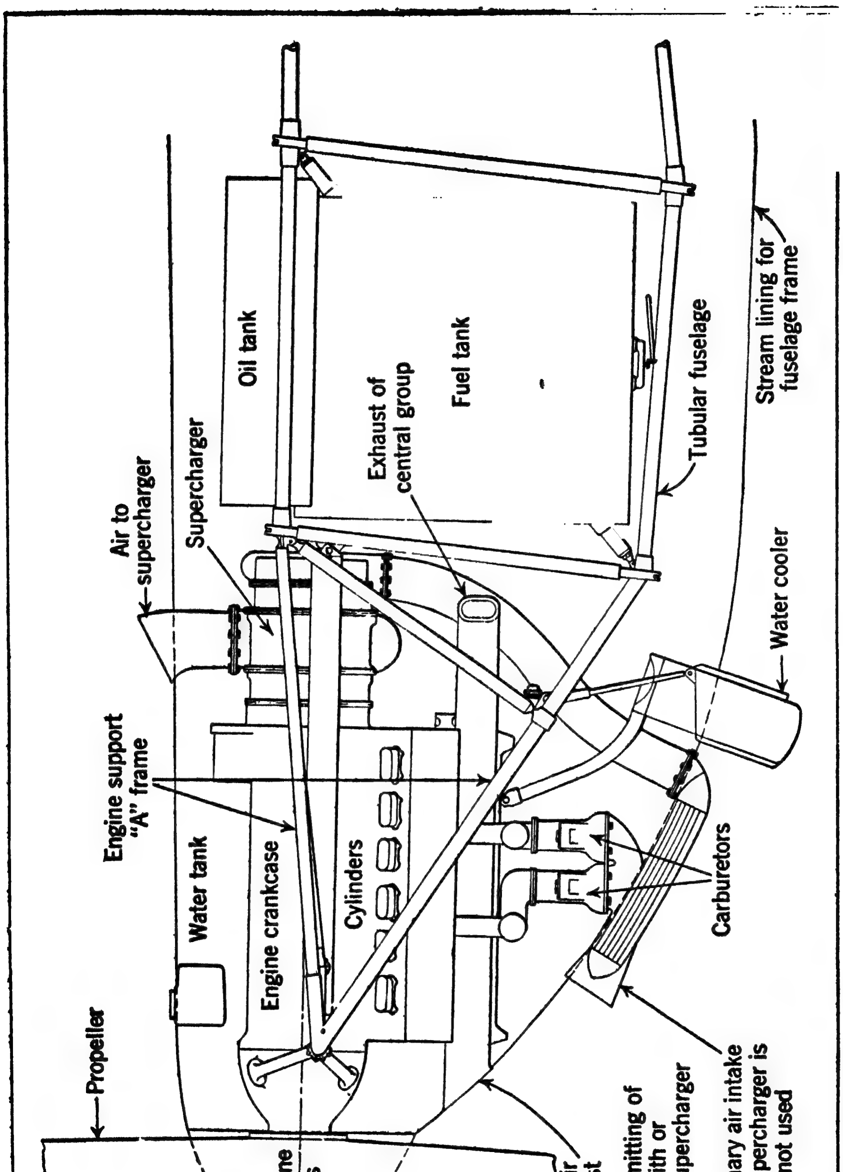


Fig. 145.—Diagram Outlining the Installation of the Farman Eighteen-Cylinder "W" Engine in the Breguet 19 Airplane Fuselage. Note the Location of the Rateau Supercharger and Arrangement of Carburetors, so Engine May be Operated with or without Supercharger.

in manifold depression at high crank-speeds. This type is used on Pratt and Whitney Wasp and Bristol Jupiter engines and on the Wright Cyclone. To illustrate the advantages to be gained by supercharging, a standard Navy shipboard observation-airplane of the UO1 type fitted with a J4 engine was equipped with a Roots blower and, as a result, its rate of climb was increased materially at all altitudes above sea level and its service ceiling was exactly doubled.

Farman Supercharged Engine.—With propeller drive reduction ratios of 2 to 2.2 the natural tendency is to increase the number of revolutions, and the opinion of many European engineers is that for power outputs of 500, with twelve or 24 cylinders, aircraft engines will run at 3,000 r.p.m. One of the most outstanding engines of this type is the new Farman 600-700 horsepower eighteen-cylinder inverted, with cylinders in Alpax metal, now made not only with a geared down propeller but with a Rateau mechanical supercharger. This engine runs at 2,800 r.p.m. and weighs 700 pounds and is shown installed in a Breguet airplane fuselage at Fig. 145 and the engine with supercharger installed is shown at Fig. 145 A.

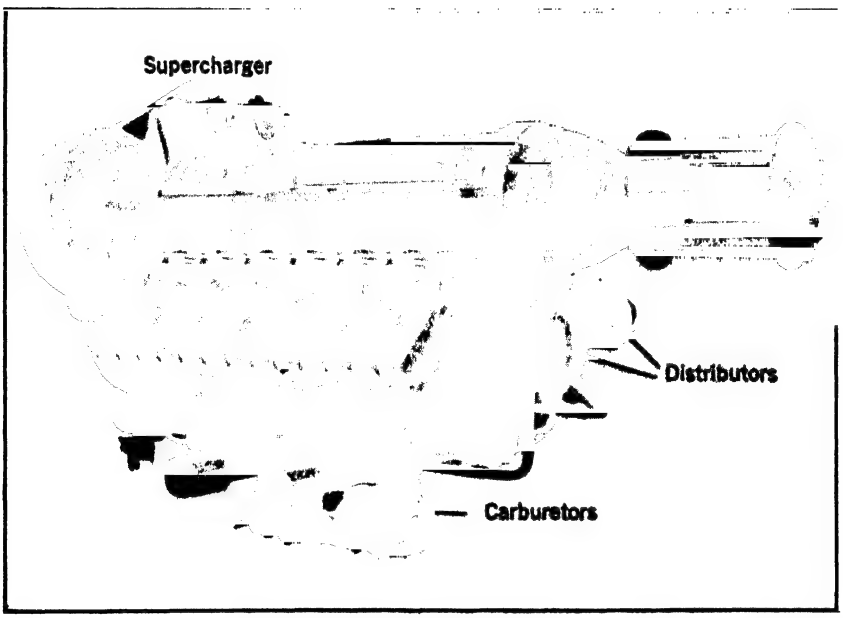


Fig. 145A.—The Farman Eighteen-Cylinder 700 Horsepower Inverted Motor with Supercharger Built Integrally Into the Engine.

Several French engines are now shown with the Rateau mechanical supercharger, which appears to have been developed by this firm in conjunction with the Farman company. Rateau produced a turbo-compressor, making use of the exhaust gases, in 1917, its first application being on an engine of 175 horsepower and later to 300 and 450 horsepower engines, on which it maintained the ground pressure up to altitudes of about 17,700 feet. The mechanical compressor now introduced maintains atmospheric pressure in the intake pipes up to altitudes of 18,000 feet. The horsepower

curves of the Farman engine with and without the Rateau compressor are shown at Fig. 146. Two models have been produced up to the present, for engines of 450-550 and 600-750 horsepower, respectively.

The centrifugal compressor is brought into engagement by means of a clutch operated by a lever, and is geared up in relation to the engine, although the ratio is not stated. The body of the compressor is in two parts in Alpax metal, the parts being united by twelve bolts so the angular location of the inlet in relation to the outlet can be raised by steps of 30

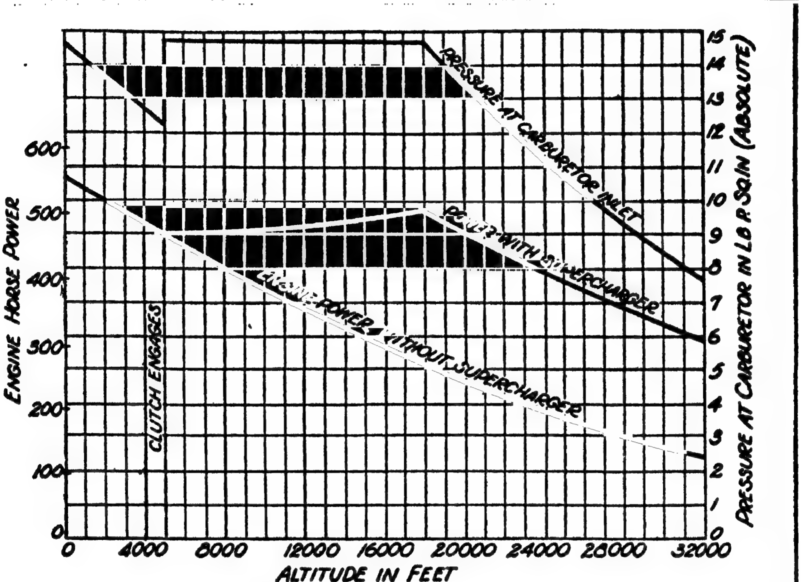


Fig. 146.—Horsepower Curves of Farman Engine with and without Rateau Compressor.

degrees. The total weight of the compressor is 110 pounds. While it is possible to adapt a compressor of this type to most engines, there would be advantage in standardizing so that any engine could receive a supercharger as desired.

Two carburetors are fitted as shown at Fig. 145 A, one providing mixture to the nine rear cylinders, the other to the nine forward cylinders. Experiments have been carried out with a Rateau supercharger geared up in relation to the engine with a ratio of one to seven. A single disc clutch into engagement by a very light spring; as speed is increased centrifugal assures connection between supercharger and engine and is first brought weights on the clutch assure a positive drive. Experiments made up to the present show that the supercharger is capable of restoring the ground level horsepower at an altitude of 18,000 feet. Tests have been carried out at the French Government Laboratory at Chalais-Meudon, and one engine has been accepted after a 50-hour test. An engine of 500 hp. maintaining its full power up to 19,000 ft. would increase its speed from 130 m.p.h. at ground level to 165 m.p.h., and it is claimed that if Lindbergh's

machine had been supercharged it could have flown to Paris in 18 hours.

Pressure or Suction Supercharging.—Two methods of supercharging are employed; first, the pressure type system as illustrated in simple diagram at Fig. 147 A and second, the suction type system illustrated in Fig. 147 B. In the pressure type system the carburetors are located between the supercharger and engine and are operating under supercharger pres-

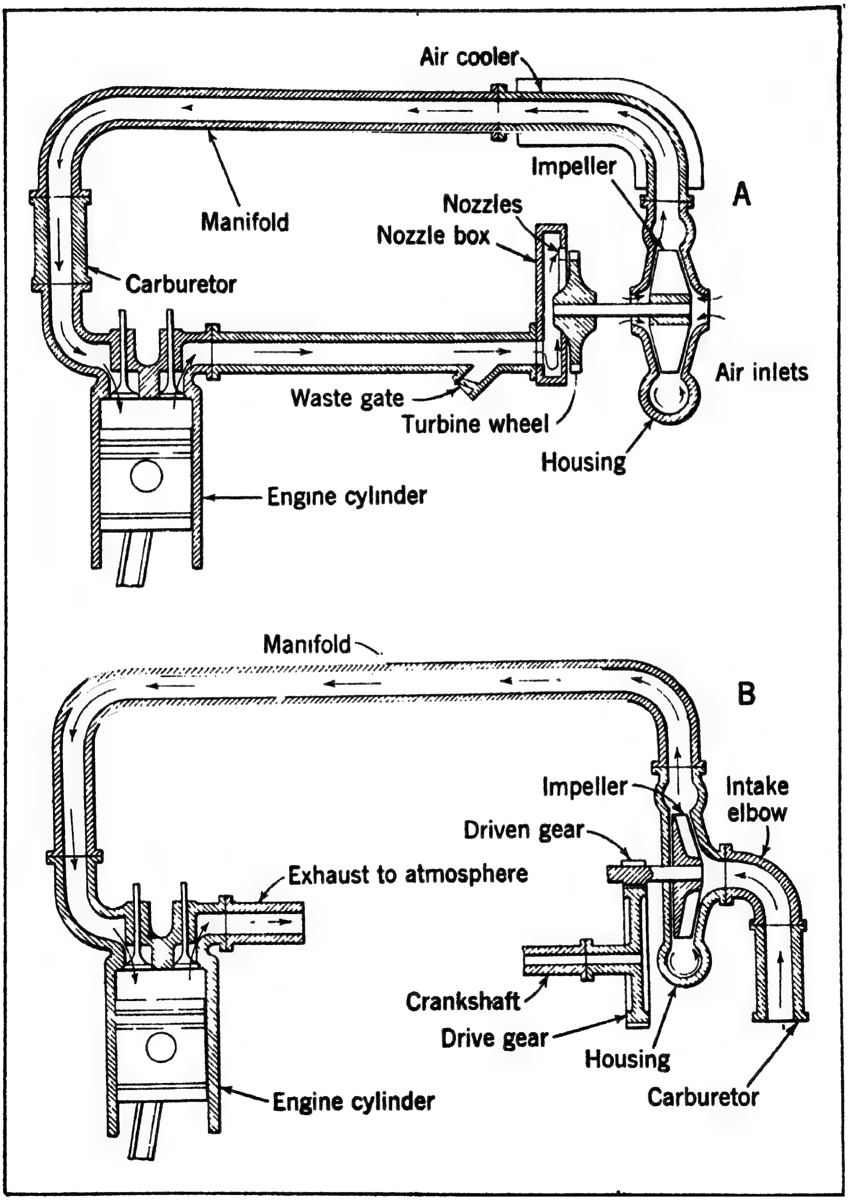


Fig. 147.—Simplified Diagrams Showing Methods of Driving Centrifugal Supercharger. A—By Exhaust Gas Driven Turbine. B—By Mechanical Gearing.

sure at practically all times; while in the case of the suction type system the carburetors are located at the supercharger air inlet and are operating under normal atmospheric pressure. Both of these systems have their particular merits and the use of either one is practically dependent upon its adaptability to the installation in which it will be used. The methods of controlling the supercharger output varies somewhat in the different types. In the turbine type the waste gate controls the volume of exhaust gases impinging on the turbine blades. By closing the gate the speed of the turbine and hence the amount of supercharging is increased. In the displacement type and direct driven centrifugal units the control is by means of a gate on the discharge side of the compressor, while in the case of the suction type the carburetor throttle acts as a control. The amount of supercharging is measured by a sealed altimeter or pressure gauge located in the cockpit and which indicates the engine altitude or pressure. With the airplane flying at 20,000 feet and the indicating instrument reading zero or sea-level pressure it would mean that the carburetors were getting air at sea-level density, and that the engine was operating under practically the same conditions that it would at sea level.

As a nonsupercharged airplane ascends, the engine power and the power required to drive the propeller decrease in almost the same ratio. At a height of 20,000 feet, the propeller revolutions are little lower than they would be at the ground, while the power developed by the engine has been reduced to approximately 40 per cent of its normal output. With the supercharged engine, where the power is practically constant, the propeller tends to speed up with altitude and allows the engine to race. The variable pitch propeller, in any form, has not yet been perfected to a point where it can be used as standard equipment. Therefore, some compromise must be made by using a fixed pitch propeller. If a propeller is used which will allow the engine to develop approximately normal revolutions at the ground, the engine will over-speed at altitude. A larger propeller is therefore used, designed to give normal revolutions at some such altitude as 20,000 feet. The engine revolutions at the ground will be from 100 to 250 r.p.m. less than the rated r.p.m. of the engine, and the propeller r.p.m. will increase with altitude until the rated r.p.m. of the engine is reached at approximately the rated altitude of the supercharger. In adapting a supercharger to an airplane a considerable number of other details must also be considered such as the cooling system, carburetors, fuel systems, sparkplugs and ignition; also provision for supplying the pilot and crew with oxygen.

The Materiel Division of the Air Corps has standardized more or less on two types of carburetor: the suction-type and the pressure-type. The terms, of course, merely tell the location of the carburetor relative to the supercharger. F. G. Shoemaker, of the Materiel Division, who did a great deal of work on superchargers, outlined the relative advantages and disadvantages of the two locations. In the suction-type, formation of ice in the carburetor may increase the restriction, or the ice may become loose and go into the high-speed impeller, causing chipping. This is particularly noticeable when using benzol. A third disadvantage is the large area of the inlet manifold and of the supercharger itself that must be wetted by

the accelerating charge.

The outstanding advantages of the pressure-type carburetor are the small charge of fuel and air in the inlet manifold, which causes very little injury in case of backfiring, and the lack of difficulty with ice formation, because the temperature rise in the supercharger makes the air temperature high enough so that ice will not form. Under these conditions the maximum temperature difference is obtained for intercooling, which is important with high compression-ratios, because of the great difference in temperature between the air inside and that outside the intercooler. The disadvantages are the necessity for a pressure fuel-feed to the carburetor and for a separate control-valve for the supercharger. An additional difficulty is that the carburetor must be specially built, that is, tightly sealed throughout and internally vented. Great care must be taken to use packing around the throttle and mixture-control shafts because, without it, as the altitude increases the pressure difference will force a combustible charge out around the throttle shafts and impair the action of the mixture control.

Air Corps Supercharger Development.—In general, the results of improved airplane performance may be classed in the following manner:

a. Increase in airplane ceiling by 50 per cent to 75 per cent.

b. Increase in speed and maneuverability at high altitude for pursuit airplanes.

c. Makes possible two-place fighters and photographic ships for high altitude duty.

d. Makes possible high altitude cruising for offensive bombing squadrons.

e. Will enable commercial airplanes to cruise at high altitudes and make better speeds, because of lessened air resistance. Advantages b, c and d apply to military airplanes more than to commercial types.

The investigation and development of air compressors or superchargers by the U. S. Army Air Corps is being carried on along two lines: namely, the centrifugal type (including both the turbine and gear driven units) and the Roots or rotary displacement type.

In the centrifugal type the compressing elements are practically the same for both the turbine and geared types, their primary differences being in the method of driving. In the case of the turbine driven centrifugal compressor the potential energy of the exhaust gases is controlled and made available for power to drive the turbine as in the installation shown at Fig. 151 which shows a Liberty-12 engine used in experimental work by the Army Air Corps equipped with a side type exhaust driven turbine supercharger. A Curtiss Hawk airplane, equipped with a supercharged engine is shown at Fig. 152. The device mounted ahead of the supercharger is the air-cooler. In the case of the geared type the power to drive the supercharger is taken directly from the engine crankshaft; the engine connection is usually made by means of a clutch or flexible coupling. The Roots supercharger is of the displacement type and by referring to Fig. 143 A the principle of operation of this type may be seen; the power for this type of unit is taken directly from the engine crankshaft. The centrifugal type of compressor is essentially a high-speed machine

and although the normal speed rating is in the neighborhood of 23,000 r.p.m., speeds of 35,000 to 40,000 r.p.m. have been obtained. Its extreme simplicity and compactness is remarkable when it is considered that the units recently developed are capable of handling an inlet air volume considerably in excess of 1,600 cubic feet of air per minute. The rotary compressor, operated at speeds up to 6,000 r.p.m., is slightly heavier than the centrifugal type and its reliability is primarily due to low r.p.m. The salient features of both types are reliability, low weight, flexibility and adaptability. The construction is comparable to all standard aircraft practice, requiring the maximum unit strength at the lowest possible weight. This is accomplished by using aluminum, duralumin and magnesium metals wherever possible. The use of these metals has permitted design and construction in which the total weight of the compressing units has been reduced to between 65 and 85 pounds as compared to approximately three tons for a commercial centrifugal compressor of the same displacement built for shop work and designed to operate at lower speeds. Of course, service life is sacrificed to attain extreme high speeds.

Efficiency of Centrifugal Superchargers.—Dr. Sanford A. Moss, of the General Electric Company who has done a great amount of development work on turbo-superchargers, recently discussed some of the features of this type before the S. A. E. He said:

"In the early days it was considered ridiculous to suppose that a mere pinwheel, rotating in a casing, could have any appreciable influence on engine performance. Only a few weeks ago I was seriously assured by an aviation engineer that, even though a centrifugal supercharger maintained a given pressure in the intake manifold of an engine, it was not positive, and that a positive-displacement machine, maintaining exactly the same pressure in the intake manifold, would be much more effective in forcing air into the engine. Of course, the fact is that a given manifold pressure will result in a given amount of supercharging, regardless of how the pressure is produced. Another fallacy which has been prevalent is that the impeller or 'fan' of a centrifugal supercharger 'fans' the air and produces a serious temperature rise, so that, even though a centrifugal machine had the same efficiency as a positive-displacement machine, the temperature rise would be much greater."

The figure of 70 per cent which is used for the efficiency of a centrifugal supercharger is a good average for a machine under actual conditions of operation on an airplane engine, if properly designed for these conditions. The various blade-shapes and angles and widths, the diffuser proportions and all the details must be carefully arranged. If this is not done, the efficiency may have any lower value whatever. Illicit comparisons have been made between a poorly designed fan and a so-called positive-displacement machine operated at such a low pressure and speed that the efficiency was much higher than would occur in actual operation on an airplane.

The word "positive" has been supposed by people who really should know better to have some magical effect in getting air into an engine cylinder which could not be accomplished by a centrifugal supercharger

with the same pressure and even better efficiency and, hence, lower temperature-rise.

In making the comparisons about a Roots-type supercharger at sea level, it has been stated by Messrs. Chenoweth and Jones that they thought the power would be virtually zero at sea level. Dr. Moss does not agree with that opinion because a positive-displacement Roots-type supercharger has a considerable task in forcing the air in and out of the lobes and through the openings at the high speed used in an airplane installation, so that a Roots-type supercharger, even though it gave no pressure near sea level, still would consume a considerable amount of power. No data about this have been published; so one guess is as good as another as to whether that power is appreciable or not.

Dr. Moss states that, at altitudes of from 12,000 to 16,000 ft., the use of a centrifugal supercharger without an air-cooler practically doubles the power of an airplane engine. This is an important gain, which justifies the use of such a supercharger in most aviation engines. It is to be noted that a geared centrifugal supercharger operates at full speed all the time, even at low altitudes when very little of the supercharging effect is needed. The test data show that the loss of engine power, due to this, is comparatively small. Flight tests of airplanes, with engines arranged for supercharging to considerable altitudes, show very slight loss of speed at sea level compared with unsupercharged planes. A large part of this slight loss is to be ascribed to the fact that a supercharged plane requires a somewhat larger propeller than an unsupercharged one, so that the engine speed is decreased somewhat.

Proposals have been made to use clutches or bypass valves to unload a centrifugal supercharger, but the loss is so slight that the complication has never been warranted. Of course, the supercharger easily increases the engine power by the amount necessary to balance this loss. The unused supercharging effect at small altitudes also gives a reserve whereby the engine can be made to give more than normal sea-level power as an emergency matter for short intervals. This, of course, requires an engine that can withstand such emergency increase of power without risk.

A marked advantage of the centrifugal supercharger lies in the vaporization of the mixture. The great gain in this respect requires more than mere mention. There is an appreciable improvement in starting characteristics, uniformity of firing, uniformity of distribution, and similar matters when gasoline is passed through a centrifugal supercharger by use of a carburetor located at the supercharger inlet or operating on the suction instead of the pressure supercharging system. It is mentioned that a carburetor drop of about 1.5 to 2.0 in. of mercury is common. However, it is believed that the vaporization secured in the centrifugal supercharger will allow the use of lower carburetor velocity and so diminish this drop. The centrifugal supercharger vaporizes practically all the gasoline before the cylinder is reached, with the resulting temperature-drop and increase of weight of charge in a given volume. With the unsupercharged engine there is probably some unevaporated gasoline at the beginning of the compression stroke. S. W. Sparrow, when at the Bureau of Standards, computed that the vaporization of gasoline gives a 36-degree Fahrenheit temperature-decrease.

Blowers for Charge Distribution.—The Bristol Jupiter air-cooled engine of 146 by 190 mm. bore and stroke is made with both a supercharger and a reducing gear, the latter being the planetary type. The blower is of the high speed centrifugal type embodying various patented features which are claimed to eliminate the inertia problems inherent with this type of mechanism. A system of slipping clutches insures that the impeller is protected from shock loads, the torque in the blower drive being

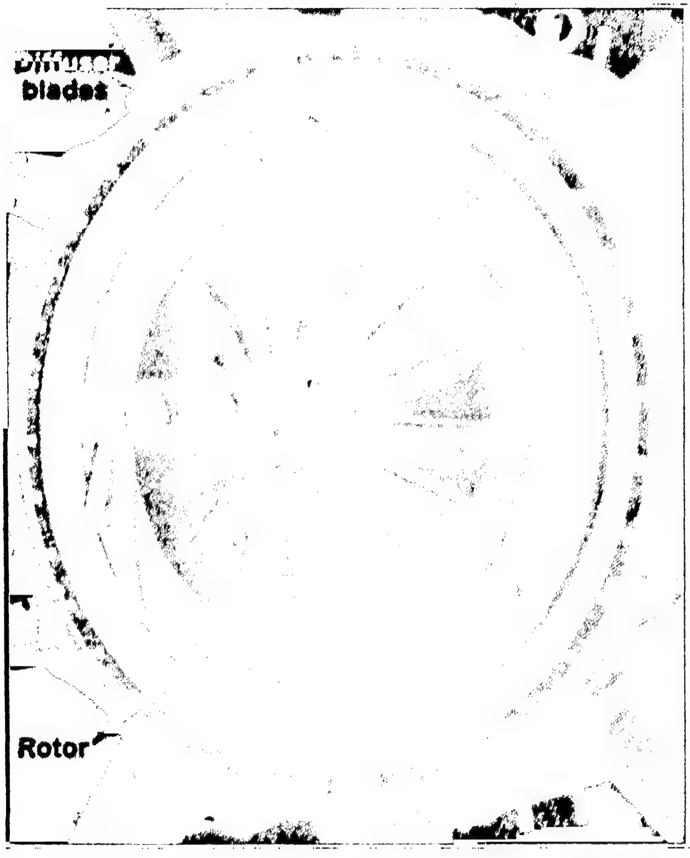


Fig. 148.—The Jupiter Gear-Driven Supercharger with Cover Removed to Show Impeller, which Rotates at High Speed, and Fixed Diffuser Vanes which do not Rotate.

practically constant. The unit shown at Fig. 148 is mounted immediately behind the rear wall of the crankcase, concentric with and driven from the tail end of the crankshaft, around which it revolves. In order to retain simplicity of the standard type of carburetor and controls, the triplex carburetor is mounted on the intake side of the blower, the mixture being drawn axially into the impeller and discharged radially via a fixed diffuser into the annular diffusion chamber, which replaces the induction spiral.

The Pratt and Whitney Wasp engine is characterized by its light weight per horsepower, clean design, simplicity of construction, remarkable ease of maintenance, and excellent operating-qualities. It is shown

in transverse section at Fig. 149, the important parts being clearly indicated. On an extremely conservative rating, this engine develops more power than the Liberty-12 engine on its maximum rating. The installed weight of the Wasp is approximately one-half that of the Liberty-12 engine. The magneto couplings are the only exposed moving parts on the Wasp engine, and the symmetry and form of the engine make the

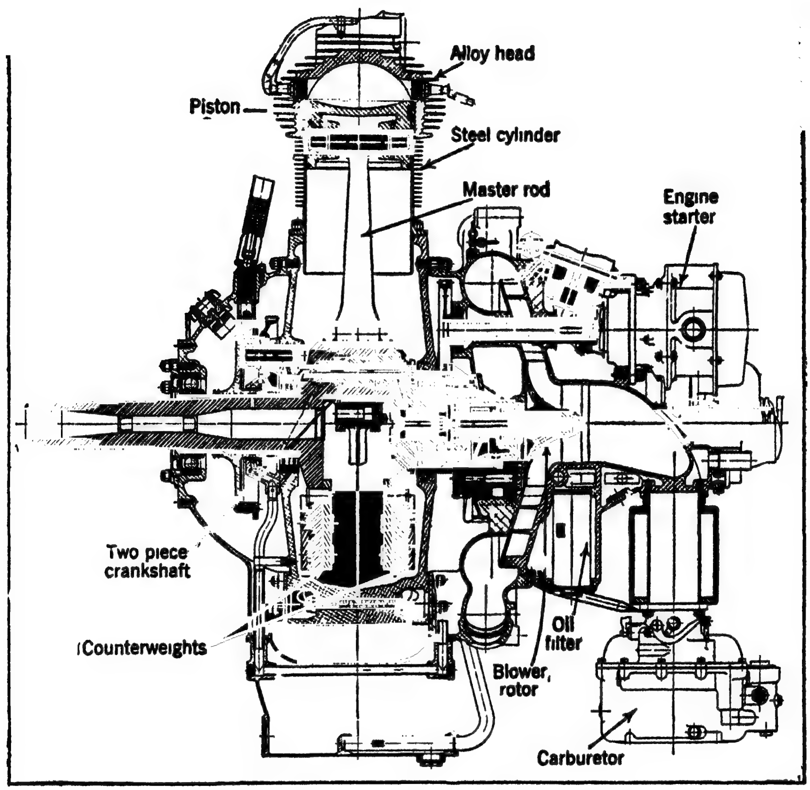


Fig. 149.—Sectional Diagram of Pratt & Whitney "Wasp" Engine Showing Location of Gear-Driven Centrifugal Supercharger at Anti-Propeller End of Crankcase.

cowling arrangement simple, efficient and generally pleasing. All accessories are grouped in the rear section so that they are enclosed in the first cowled-in bay of the fuselage, where they are protected and also are convenient of access as shown at Fig. 150 which outlines the engine on the Vought Corsair detachable engine installation unit.

To secure the best possible distribution and high volumetric efficiency, a General Electric centrifugal compressor is fitted in a thin section immediately behind the crankcase. It is for rotary distribution only and is not a supercharger. Its slight supercharging effect is just sufficient to overcome manifold depression at full throttle. The impeller is spur-gear driven from the rear end of the crankshaft at five times engine-speed. It draws carbureted air directly from a Stromberg carburetor and delivers

it, through a diffuser and the individual intake-pipes, to the valves.

The rear cover contains the entire accessory group. The accessories are admirably laid out in a manner which provides compactness and accessibility. The accessory group can be passed through the mounting circle of the engine, which greatly facilitates the removal of the engine from the airplane. The blower section which carries the mounting lugs and the rear cover can be left in place and a new power-unit can be bolted-on without disconnecting any parts. The oil-pump strainers are built into the casting while the magnetos, the carburetor, the starter, the fuel pump, and the generator, if a generator is used, are mounted on flanges.

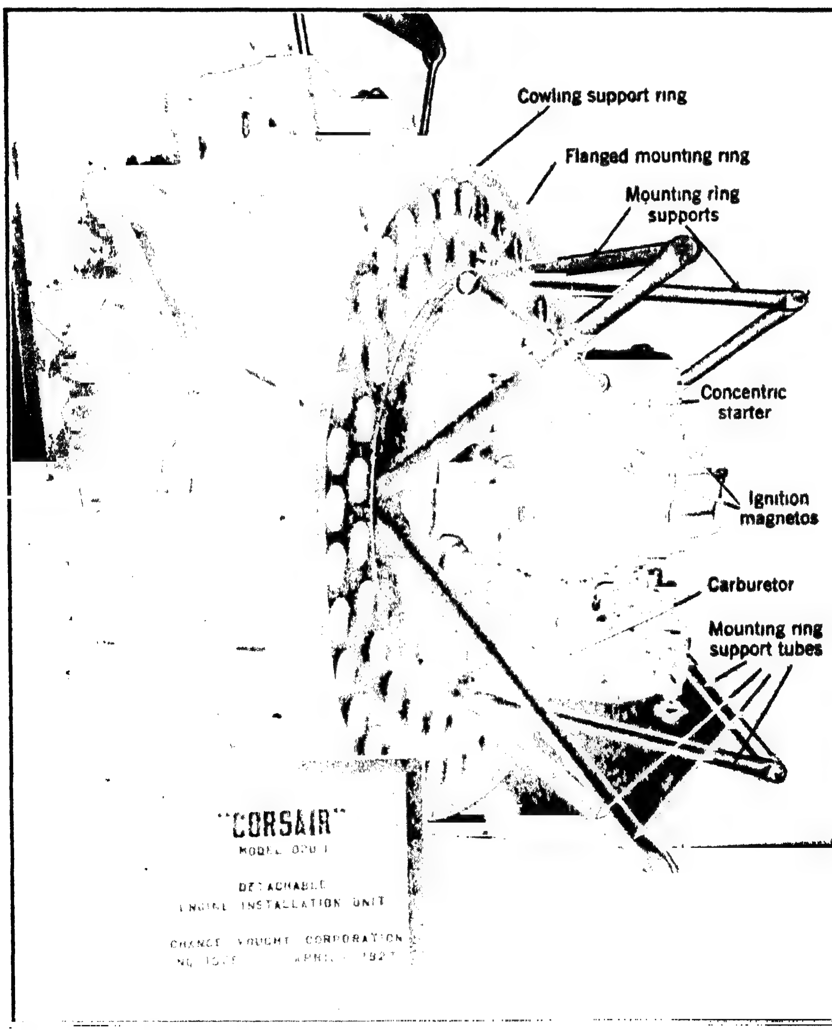


Fig. 150.—Pratt & Whitney "Wasp" Engine Installed on Vought-Corsair Detachable Engine Mounting Showing how all Accessories are Grouped at Anti-Propeller End of the Engine.

Superchargers Aid Scavenging.—Another item of advantage with superchargers is that, with proper valve-timing, an appreciable amount of scavenging can be obtained; that is, a large portion of the products of combustion occupying the clearance space can be driven out and the clearance space itself filled with fresh charge, due to the difference between the supercharger pressure and the exhaust pressure, if the exhaust and the inlet valves are opened at the proper times. Computations of this effect have been made by others on the assumption that the entire clearance space is exactly filled with fresh charge, instead of with products of combustion as in the unsupercharged engine. An important factor that merits consideration is the supercharging of aviation engines at sea level. The centrifugal gear-driven supercharger, if opened up near sea level so as to give a pressure in the intake manifold greater than standard sea-level pressure, will give a greatly increased amount of power. The engine of course must be arranged to withstand this increase in power. Arrangements for accomplishing this are now in progress and will no doubt be used commercially in the near future. When such an engine is available, centrifugal superchargers will be installed with somewhat higher gear-ratios than are now contemplated. For flying near sea level the supercharger will be partly opened up so as to give whatever increased pressure in the intake manifold the engine can safely take care of. As the plane climbs to higher altitudes the engine power will be maintained at the safe value at sea level by successively opening up the supercharger. In aircraft-engine work, however, the conditions for supercharging are ideal, because the engines operate in an air density that is considerably less than that at sea level; they also secure the benefits of the low temperatures at the higher altitudes. This is especially beneficial insofar as turbo-supercharger operation and engine temperatures are concerned.

Practical Value of Superchargers.—In discussing the subject of superchargers in aircraft work, A. L. Berger of the Supercharger Dept., Wright Field, Dayton, Ohio, brings out some interesting experiences obtained by the Army Air Corps. At the time of the inception of the supercharger for aircraft engines, nearly all known types of compressor were considered, but the turbo type seemed to be best adapted to aircraft, probably because the kinetic energy available in the exhaust gases could be utilized, and the necessity for taking power from the crankshaft for operating the supercharger could be avoided. Some idea of the results achieved from supercharging aircraft engines may be obtained when one considers that a 400-hp. Liberty engine at 20,000-ft. altitude without a supercharger develops approximately only 195 hp. and that, by supercharging the engine to sea-level pressure at that altitude, practically normal sea-level horsepower is again made available. The hardest problem encountered in the development of the turbo-supercharger was that of securing a satisfactory metal for the turbine buckets. When one considers that the turbines operate in red-hot exhaust-gases, the temperatures of which are from 1,200 to 1,500 degrees Fahrenheit, he can realize the difficulties of the problem. After six or seven years the Army Air Corps finally developed a metal for the turbine buckets that has proved very successful. While the Air Corps has standardized on turbo-superchargers for aircraft work, that does not close the

field to other types of supercharger. They are considering doing some work with vane-type blowers, some designs of which seem to offer possibilities.

As an illustration of the performance of an airplane equipped with a supercharger, Mr. Berger cites the present Air Corps standard pursuit-airplane, which, when equipped with an unsupercharged engine, has a time of climb to 20,000 feet of approximately 26 minutes; but, when the engine is equipped with a supercharger, the time of climb to the same altitude has been reduced to less than twelve minutes. It is likely that in the near future altitudes of 20,000 feet will be reached in seven or eight minutes. After considerable experimenting with the turbo-type supercharger, it was believed possible to connect a gear-driven centrifugal supercharger directly

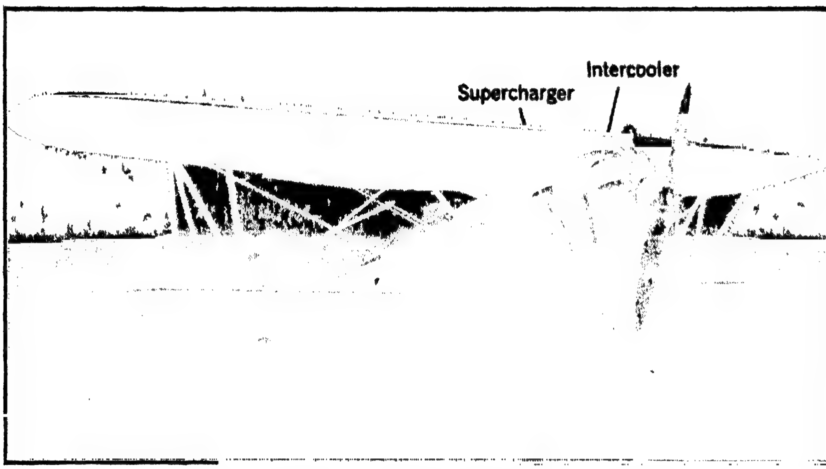


Fig. 152.—Official Photograph from U. S. Army Air Corps Showing Installation of Powerplant Employing Experimental Side Type Supercharger on Curtiss Military Airplane.

to the rear of the engine. There was, however, a rather grave question whether a satisfactory connection could be made, because of the oscillation of the crankshaft. Later developments have shown this skepticism to be well founded.

A great many types of couplings and drives have been tested, but none has proved entirely successful. It is probable, however, that this difficulty can be largely overcome by building the supercharger directly into the engine. Some work has also been done with the Roots-type supercharger, as developed by the National Advisory Committee for Aeronautics. The tests of this type to date, however, have not been conclusive, as a number of mechanical failures have been encountered. In general, mechanical difficulties have been the cause of the greatest trouble, irrespective of the type of supercharger. These are probably more serious than those of superchargers used with automobile engines, because superchargers for aircraft engines must be constructed of the very lightest material. Superchargers used with aircraft engines of 400 to 500 horsepower weigh less than 100 pounds. The light construction increases considerably the possibility of mechanical trouble.

With regard to supercharger air-discharge temperature, the final temperature should be approximately the same for the same compression-pressures, whether the supercharger is a turbo, geared centrifugal, rotary or vane-type compressor. That of the centrifugal type probably is slightly less than that of the others at the higher operating speeds. Intercoolers are essential for cooling the air, especially when high mean effective engine-pressures and high supercharger-pressures are maintained. The power requirement for supercharging at seven-pounds per square inch pressure-difference is about four horsepower per 100 cubic inches of piston-displacement at 3,000-r.p.m. engine-speed. In return for this we receive a

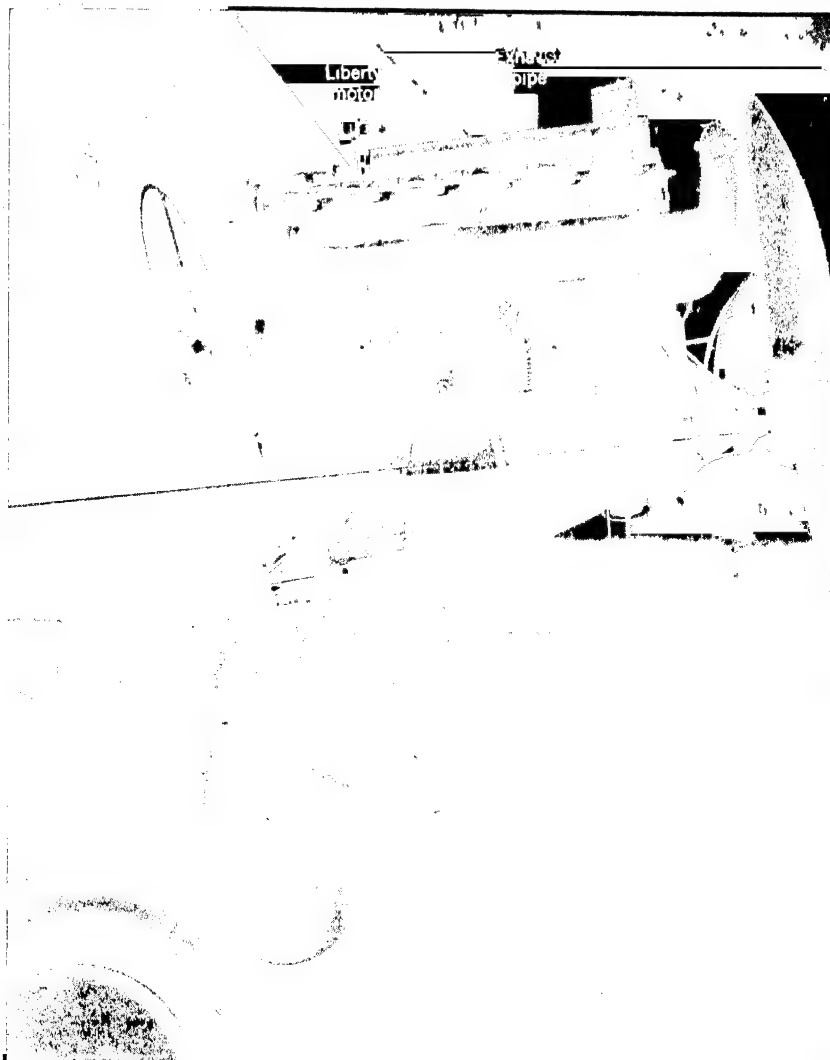


Fig. 151.—Official Photograph Supplied by the U. S. Army Air Corps Showing Side Type Supercharger on Liberty Engine.

net increase in percentage of power 1.8 times the increase in percentage of volumetric efficiency. At 3,000 r.p.m., few engines have more than 70-per cent volumetric efficiency at sea level. A volumetric efficiency of 100 per cent can be obtained with supercharging, or a net increase in power of 55 per cent.

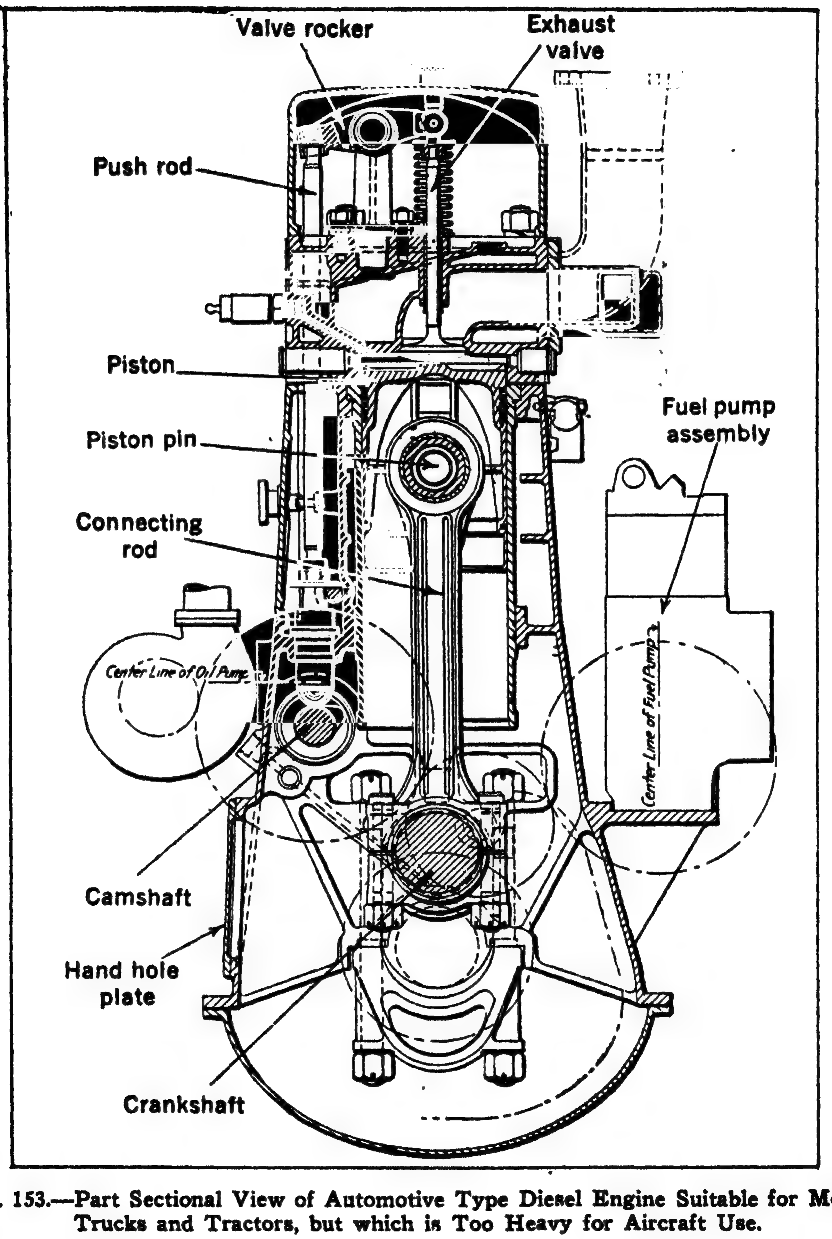


Fig. 153.—Part Sectional View of Automotive Type Diesel Engine Suitable for Motor Trucks and Tractors, but which is Too Heavy for Aircraft Use.

Automotive Diesel Engines.—The automotive Diesel engine that has been built for motor truck use would have possibilities in aircraft applications if the weight could be reduced below the twenty pounds per horsepower that seems to be the low limit at the present time. An engine intended for automotive use is shown at Fig. 153. It has four cylinders of six inches bore and eight inches stroke and light materials are used at various points to reduce the weight. The engine was described by R. J. Broege in the February issue of the *S. A. E. Journal.* The general construction is apparent by inspecting the transverse sectional view. Careful consideration has been given to accessibility of all working parts for ease of inspection and servicing. The integral crankcase and cylinder-housing with walls between cylinders from top to bottom gives a very rigid construction and permits the use of much thinner sections, which tends to decrease the weight of the engine. The crankcase and cylinder-housing are of boxed construction cast-in-block integrally for any number of cylinders. This produces a rigid and light design and causes the engine to operate very smoothly. The cylinder proper comprises a gray-iron inserted sleeve which is readily renewable. The main babbitt-lined bearings are suspended in the crankcase and secured by inserted bolts, there being a bearing between each two cylinders. The oil-pan is of aluminum to secure lightness so that it can be easily dropped, as in most automotive-type gasoline-engines, for bearing adjustment. Provision has also been made for removing the pistons and their rods from the bottom; but they can also be removed through the top if desired. Hand-holes are provided on the side of the crankcase for connecting-rod-bearing adjustment.

Either electric starters or a two-cylinder air-cooled double opposed-piston gasoline-engine can be used for starting purposes, either system being arranged with a gear drive automatically engaging the ring-gear on the flywheel. To aid the initial turning-over of the engine, a manually controlled compression release has been provided. After the engine is spinning, compression can be engaged on one cylinder until it fires, after which all cylinders can be engaged. The valves are located in the cylinder-heads and are operated by rocker-arms and roller push rods. Valve-tappet adjustments can be made conveniently by removing the cylinder-head covers, which leaves the rocker-arms in plain view without obstruction by accessories or other parts. The valve-in-head design is used to obtain the desired compression space for the most efficient pressures. The inlet-valve has an integral deflector on the head which, with the angular direction of the fuel spray, produces the necessary turbulence for the complete mixture of air and fuel that is necessary to produce clean combustion. The cylinder-heads are cast in pairs to secure lightness and to facilitate easy removal, and are protected by safety-valves.

The air-inlet pipe has a pre-heater attached to aid in starting under extremely cold conditions and is arranged for the application of an air-filter. The exhaust pipe is large and has an outlet flange suitable for attaching a muffler. The fuel-pump is mounted on a bracket integral with the crankcase and is positive, being driven off the timing-gear train. It is a cam-actuated plunger type with a plunger and a cut-off valve for each engine cylinder. The cut-off valve and plunger are actuated by fulcrum

tappets interposed between the main-drive cams and the plunger and cut-off valve. The cut-off valve can be adjusted easily in the same manner as when adjusting ordinary valve-tappets. All moving parts are enclosed in a common housing and operated by one drive-shaft having integral drive-cams.

The engine-speed actually is controlled by the governor, which is built into the fuel pump and operates the controlling, or cut-off, valves. These valves can also be controlled manually if desired. The speed of the engine is increased or decreased by the point of the fuel cut-off and the duration of injection, depending upon the plunger and valve position in relation to piston position and speed. The two controls are the only ones for operating the engine and correspond very closely to the spark and gasoline controls of the gasoline-engine. The fuel-injection advance-lever location can be determined and located readily for a wide range of speed, and the engine is then controlled by the fuel cut-off lever. An engine of this type may be run at speeds up to 1,200 r.p.m. It was tested with three different grades of fuel oil. The engine burned each of these fuels equally well during all tests. In addition to the light weight, the small space required and its simplicity as compared with the conventional air-compressor-injection Diesel-engines, this engine has many other points of merit when compared with other types of engine.

Advantages of Automotive Diesel Engines.—The advantages of the Diesel automotive type have been summed up as follows by Mr. Broege:

Compared with Gasoline Engines

(1) It derives its fuel from the largest supply of the cheaper and non-volatile hydrocarbons.
(2) It has a lower fuel-consumption per horsepower, and especially so under variable load-conditions.
(3) The automatic ignition eliminates accessories and drives such as are required for magneto or distributors, which results in greater reliability.
(4) It simplifies or eliminates possible difficulties which might arise from additional and complicated equipment.

Compared with Fuel-Oil-Vaporizing Engines

(1) Auxiliary fuels for starting are eliminated.
(2) It is always ready for starting.
(3) It has improved adaptabilities for automotive purposes.
(4) There is no contamination of lubricating oil by fuel oil.

Compared with Engines of Ignition-Chamber Type

(1) The cylinder-head construction is more simple, especially considering provision for starting.
(2) The possibility of requiring auxiliary fuel for starting is avoided.
(3) Ignition cartridges or electric ignition-coils, with their necessary batteries, are eliminated.
(4) It has better adaptability to the commercial automotive trade and to any other work for which the automotive gasoline-engine is used.
(5) The operation with a cold engine or under no load is more uniform, even at low speeds.

Compared with Hot-Bulb Engines

(1) It has lower fuel-consumption, especially under heavier loads.
(2) It weighs less.
(3) The lamp for heating the hot bulb is eliminated.
(4) It is always ready for starting.

These automotive Diesel engines should in no way be compared with the commonly known slow-speed heavy Diesels of either the air-injection or solid-injection types. Mr. Broege believes it is fair to state that the automotive high-speed Diesel engine is here and has already gone through many of the stages that the present-day automotive gasoline-engine has gone through before reaching its present highly developed stage. This type of automotive Diesel engine in its present form will perform economically with cheaper fuel over a wide range of service, such as in commercial motor-trucks, tractors or in excavator machinery, equally as well as gasoline-engines and with far better fuel economy and less fire risk. It will perform equally well in any other service for which the gasoline automotive engine is used, and if its weight can be reduced it will be suitable for aircraft.

Peugeot-Junkers Type Diesel Engine.—Another type of Diesel automotive engine is shown at Fig. 154. This is built by Peugeot in France under Junkers' license and is the same type, as far as operating principles are concerned as that Junkers' is experimenting with for aircraft use. It has been recently described in *Automotive Industries* by W. F. Bradley, their European correspondent.

The Peugeot engine is built for motor truck use. Only one model is being produced at present, this being a two-cylinder vertical two-stroke, developing 45 horsepower at 1,200 r.p.m. Smaller units will be produced later. The Peugeot-Junkers differs from other Diesel truck engines at present on the market in being a two-cycle model with opposed pistons. The bore of the cylinders is 80 mm. (3.1 inches) and the combined stroke of the two pistons is 300 mm. (11.8 inches) giving a piston displacement of 183 cubic inches.

The power output is 40 horsepower at 1,000 r.p.m., increasing to 45 horsepower at 1,200 r.p.m., with a fuel consumption of 0.38 pounds per horsepower-hour. It is claimed that between 29 and 50 horsepower the specific consumption remains constant, and that the load has to be dropped below 29 horsepower, or carried beyond 50 horsepower for the consumption to exceed 0.40 pound per horsepower-hour. The cylinders and crankcase constitute a single aluminum casting, with iron liners set into the cylinder barrels. By reason of the opposed piston design there is no cylinder head. The lower piston is connected to the central one of three throws on the crankshaft and the upper piston to two lateral throws. The upper piston is double-ended, the lower portion, which carries four compression rings, operating in the 80 mm. bore cylinder, and the upper end of the skirt, which has a much bigger bore, forming the piston of the air cylinder for the scavenger pump.

The auxiliary connecting rods, which are under tension, are attached by split bearings to the crankshaft and at the upper end have a roller

bearing attachment to the cross beam carried in the piston, which latter is self-centering in the working cylinder. The lower piston is of normal construction and design, with five compression rings and two scraper rings. The crankshaft is carried in three plain bearings lubricated under pressure. By reason of the opposed piston construction, all the main stresses are absorbed by the moving parts and are not transmitted to the housing—the

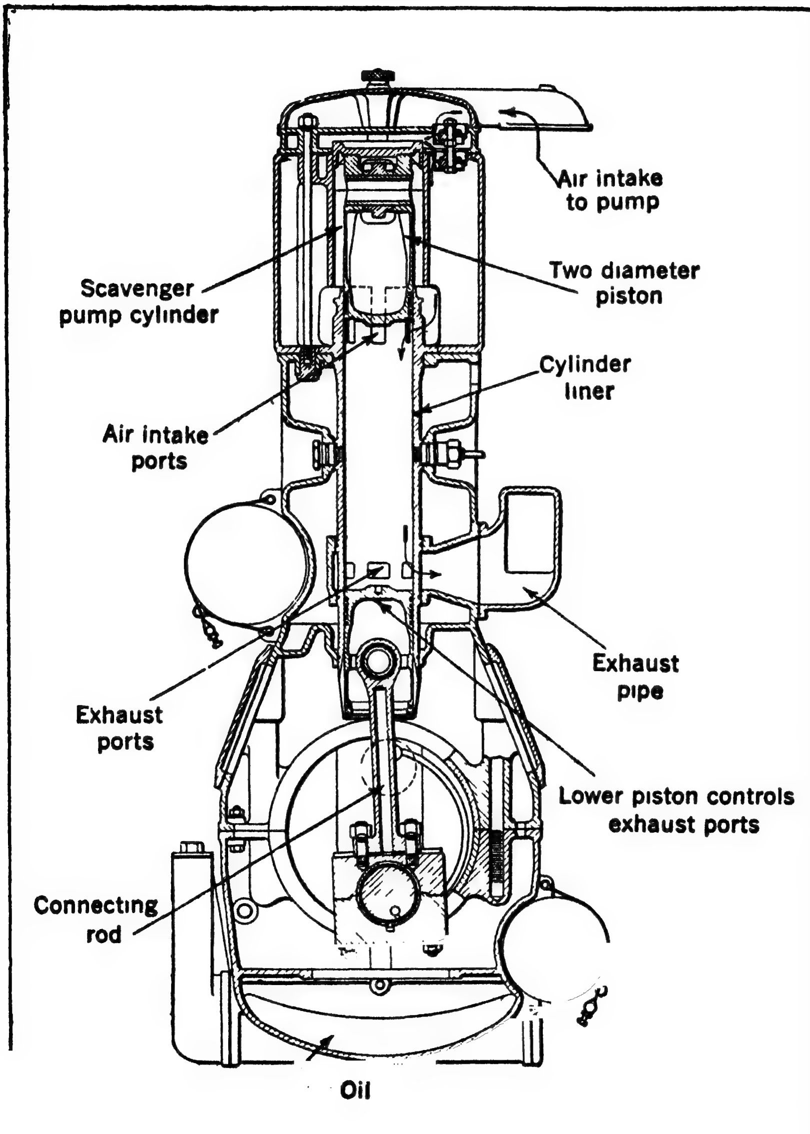


Fig. 154.—Transverse Sectional View of Peugeot-Junkers Double Piston Type, Two-Cycle Oil Injection Engine.

crankcase and the cylinder head. This makes it possible to use higher pressures than on other engines, the volumetric compression ratio being about nineteen. A higher rotational speed can be maintained with a low piston speed, and by reason of the small cylinder bore the wall surface of the combustion-chamber is reduced to a minimum.

The upper piston covers and uncovers the inlet ports, while the lower piston uncovers the exhaust ports. The stroke of the two pistons is not the same, thus making it possible to give a lead to the exhaust opening. Only pure air is admitted, this first driving out the remains of the exhaust gases and then being compressed. Fuel is injected by means of a constant-stroke fuel pump, starting about seventeen degrees before the upper dead center. The injection point is invariable, but the quantity of fuel delivered at each stroke of the pump is varied through the operation of the throttle and the accelerator.

The fuel pump, which is Junkers' own design, is fed from a constant level float chamber and is driven direct from the front end of the crankshaft. On the left-hand side of the engine, in tandem, is the water pump and the electric generator. The usual type of electric starting motor, with Bendix gear, is fitted. The radiator fan is positively driven. The weight of the engine, without flywheel, is given as 616 lb., minus accessories, water and oil which makes it much too heavy for aircraft. In operation the engine is free from smoke or any objectionable smell, and it assures a greater degree of flexibility than a normal four-stroke, four-cylinder truck engine of equivalent power and does not weigh very much more.

Fuel Injection a Problem.—A number of authorities agree with the choice of airless injection and open combustion-chamber. Because of the fewer moving parts resulting in a simpler design some prefer the two-stroke cycle, but it must be admitted that neither the art of scavenging nor that of heat dissipation is sufficiently advanced to allow building a 3,000-horsepower two-cycle engine to run at 700 r.p.m. on a basis of safely established practice as has been done by Mr. Trieber in the marine engine previously described and illustrated.

In a solid-injection engine, the most debatable question is between constant-pressure injection and the jerk pump. Among the troubles with the direct pump system are after-dripping; misses caused by air bubbles; secondary discharges from pressure waves; destructive jerks of the pump-plunger; water-hammer action in the fuel lines; varying injection-lag, due to both the compressibility of the liquid and "breathing" of the pipe; and falling off of the injection pressure and quality of atomization at low speed.

While the constant-pressure system is free from these troubles, it has disadvantages of its own: (*a*) To be effective, the injection control must be in the spray nozzle, and a mechanically operated spray nozzle is complicated; (*b*) with fuel continually under pressure in the line, even a slight leakage in the injection nozzle will cause dripping that is hard to detect, and the dripped fuel may cause undesirable explosion pressure; (*c*) metering of minute quantities of fuel by keeping the valve open for a definite length of time is not positive enough, and the distribution becomes uneven if some nozzle orifices are widened by erosion or clogged.

As a rule, high fuel-economy does not accompany high mean indicated pressure. In an engine with fourteen to one compression ratio, 185 pounds mean indicated pressure can be secured on a fuel consumption as low as 0.25 pound per horsepower-hour, but both cannot be obtained together, according to the theory developed by George A. Goodenough and John B. Baker. The lowest theoretical fuel consumption with this mean indicated pressure is about 0.41 pound per horsepower-hour. With 0.25-pound fuel consumption, the theoretically highest mean indicated pressure is 104 pounds. If the excess air is cut down to below about 50 per cent, too much efficiency is sacrificed. This is confirmed by experiments showing the highest thermal efficiency at one-half and one-quarter loads.

At speeds higher than 1,600 r.p.m. it is very difficult to secure high mechanical efficiency in a Diesel engine, largely because of the friction of rings of the number and tension required.

Experience confirms the advantages of the jerk pump, and it is stated it can be made to operate a remote automatic valve with even more precision than cam actuation. The automatic valve developed by the National Advisory Committee for Aeronautics is particularly light and simple. The spray holes for a cylinder having a bore of even only three inches need not be smaller than $\frac{1}{64}$ inch. A short stroke is advisable for high speed.

A point in connection with the characteristics of Diesel engines is brought out in an item in the *Commercial Motor* based on a communication from Wm. Beardmore & Co., builders of high-speed Diesel engines. It is stated that whereas the output of a carburetor type engine falls off appreciably when the temperature of the engine exceeds a certain value, owing to the expansion of the incoming charge and the consequent loss of volumetric efficiency, with a Diesel engine there is no such effect. Tests were carried out with a high-speed Diesel engine with the cooling water at 240 degrees Fahrenheit, the cooling system being kept under pressure, and under these conditions there was no appreciable falling off in the power. Of course, the amount of air entering the cylinder will be affected the same in the Diesel as in the carburetor engine by an increase in cylinder temperature, but the Diesel engine runs normally with a large excess of air, which is necessary in order to be able to burn the bulk of the fuel injected in the very short time available. With an increase in the temperature of the air, vaporization and combustion evidently proceed more rapidly and not so large an excess of air is needed.

Sperry Oil Engine for Aircraft.—While most American manufacturers have been content to reduce gradually the weight of their smaller oil engines to a point where they are practical for use in various forms of heavy mobile equipment, such as contractor's shovels and draglines, the work of the Sperry Gyroscope Company, under the supervision of Elmer A. Sperry, has for several years been directed to the study of a suitable heavy oil engine for aircraft. The progress of this project has been summarized in a paper by Mr. Sperry which was recently read before the Metropolitan Section of the Society of Automotive Engineers, by H. J. Scharnagel, Chief Diesel Engineer for the Sperry Company. Briefly alluding to the present heavy weight of the oil engine, the author begins his discussion by citing

the fuel economy obtainable by use of the oil engine cycle. The fuel consumption of the oil-engine, compared with the gasoline-engine is roughly on the order of two to three, so that if the engines can be placed on an equal footing as to weight per horsepower, the case in favor of the oil engine is hardly open to serious question. In view of the fact that the present most highly developed light weight Diesels have a weight per horsepower ranging from sixteen pounds to five pounds, (although in some cases lower values have been reached), the significant weight reduction needed to reach a point comparable with aero engines is quite obvious. The difficulties encountered have been, in the structural sense, largely due to the higher maximum pressures and consequently higher stresses and bearing pressures which are encountered in Diesel practice.

The method of attack pursued in the Sperry laboratory has, however, been largely confined to the evolution of a design which would permit the use of a much higher mean effective pressure than is ordinarily used. This has been proposed by the use of a high initial supercharging pressure combined with a two-stage expansion which continues the power impulse of the expanding gases through a larger arc of the crankshaft revolution than is possible in oil engines which exhaust to the outer air. Experimental results have led to the confirmation of these conclusions and have been so encouraging that the production of an aviation engine is promised in the near future.

Due to the excessive amount of air which is taken into the Diesel cylinder, the combustion-chamber in the Sperry cylinder has been enlarged to about four times the normal volume. Air at about 40 pounds pressure from a pre-compression or supercharging pump is delivered to this chamber and, on the compression stroke, fuel oil is injected shortly before top dead center. A very high maximum pressure and a mean effective pressure of 300 pounds has been achieved by this method. At the beginning of the exhaust stroke, a valve in the position of the normal four-cycle exhaust valve is opened and the gases passed into a second stage cylinder and thence, at the completion of the expansion stroke in this cylinder, are exhausted into the atmosphere. The low pressure cylinder is mounted between two high pressure cylinders whose timing is arranged so that the exhaust into the low pressure cylinder takes place during alternate cycles, thus permitting it to act alternately as the expansion cylinder for two adjacent high pressure cylinders.

If pressure in the second stage cylinder were maintained at atmospheric level, the rush of the hot gases past the exhaust valve would cause serious scoring and distortion. This trouble has been prevented by maintaining a cushioning pressure in the second stage cylinder, which permits a gradual transfer of the charge; and also by cooling the outer surface of the exhaust valve through fins which are exposed to the flow of air in the channel leading to the intake valve of the high pressure cylinder. The combination of supercharging and two-stage expansion is thus announced as the principal contribution which has been made to the problem of weight reduction, the result being approached by increasing the output of the cylinder rather than by unusual reduction of masses. The difference between the normal Diesel mean effective pressure of about 80 lb. and the

300 lb. mean effective pressure of the high pressure supercharging engine is considered sufficient to overcome the handicap of the 135 lb. pressure achieved in the average aviation engine (which is the highest obtainable before pre-ignition is encountered), by a margin sufficient to make the oil engine practical for aviation service.

The difficulties encountered by other workers in the high speed field, who have not resorted to such radical means to increase the output of the oil engine, have been largely structural and otherwise intimately associated with the fuel injection system. In a high speed Diesel the time allotted for the injection of the fuel charge is so minute that the construction of a mechanical injection device capable of delivering the fuel almost instantaneously has, to a large extent, handicapped the production of high speed oil engines. No mention is made of the fuel injection system on the Sperry engine or whether the stresses encountered in high speed operation can be taken care of by an engine structure which will be within the weight limitations imposed. A power unit, built along aeronautical lines, capable of developing 250 hp. is at present in the design stage. Four of these units will be combined in a Vee Type engine to produce 1,000 hp., the dimensions of both units being tentatively determined well within practical limits, comparable to the space required for units of equivalent horsepower now being used. If the results are obtained in the aero type oil engine which have been indicated in the experimental conclusions announced by Mr. Sperry, the development will be of the utmost value to commercial aviation, which calls for a powerplant of exceptional simplicity and unusual economy. In any event the project is worthy of the closest attention because of the highly original method of development which has been chosen.

QUESTIONS FOR REVIEW

1. What is the difference in air pressure between sea level and an altitude of 20,000 feet?
2. What effect does this have on engine power?
3. Name main types of airplane engine superchargers.
4. Outline methods of centrifugal supercharger drive.
5. What is the difference between a suction type and a pressure type supercharger?
6. What is the efficiency of a centrifugal supercharger?
7. Why are blowers sometimes used for charge distribution?
8. How does a supercharger improve performance?
9. At what speeds do centrifugal superchargers operate?
10. Describe operation of Roots type blower.

CHAPTER XIII

AVIATION ENGINE IGNITION SYSTEMS—EARLY MAGNETOS

Early Ignition Systems—Electrical Ignition Best—Fundamentals of Magnetism Outlined—Magnetic Substances—Magnetic Lines of Force—Zone of Magnetic Influence Defined—The Magnetic Circuit—How Iron and Steel is Made Magnetic—Electricity and Magnetism Closely Related—Basic Principles of Magneto Outlined—Why Magneto Must be Timed—Essential Parts of a Shuttle Armature Magneto—Transformer System Uses Low Voltage Magneto—Distribution of Secondary Current—High Tension Magnetos Are Self-Contained—Function of Make and Break—Advantages of Magneto Ignition—Requirements of Aircraft Engine Ignition Systems—Magnetos vs. Battery Ignition—Comparative Weights of Battery and Magneto Ignition—Modern Engines Require Many Sparks—Magneto Drive Important Problem—Lubrication Problem Difficult—Electrical Requirements Exacting—The Berling Magneto—Two Spark Independent Magneto—Two Spark Dual Magneto—Setting Berling Magneto—Wiring the Magneto—Berling Magneto Lubrication—Adjusting the Interrupter—Cleaning the Distributor—Locating Trouble—The Dixie Magneto—Care of Dixie Magneto—Timing of the Dixie Magneto—Robert Bosch Magnetos.

One of the most important auxiliary groups of the gasoline-engine comprising the airplane powerplant and one absolutely necessary to insure engine action is the ignition system or the method employed of kindling the compressed gas in the cylinder to produce an explosion and useful power. The ignition system has been fully as well developed as other parts of the engine, and at the present time practically all ignition systems follow principles which have become standard through wide acceptance.

Early Ignition Systems.—During the early stages of development of the gasoline-engine various methods of exploding the charge of combustible gas in the cylinder were employed. On some of the earliest engines a flame burned close to the cylinder-head, and at the proper time for ignition a slide or valve moved to provide an opening which permitted the flame to ignite the gas back of the piston. This system was practical only on the primitive form of gas-engines in which the charge was not compressed before ignition. Later, when it was found desirable to compress the gas a certain degree before exploding it, an incandescent platinum tube in the combustion-chamber, which was kept in a heated condition by a flame burning in it, exploded the gas. The naked flame was not suitable in this application because when the slide was opened to provide communication between the flame and the gas the compressed charge escaped from the cylinder with enough pressure to blow out the flame at times and thus cause irregular ignition. When the flame was housed in a platinum tube it was protected from the direct action of the gas, and as long as the tube was maintained at the proper point of incandescence regular ignition was obtained. Diesel and other engineers utilized the property of gases firing themselves if compressed to a sufficient degree, while others depended upon the heat stored in the cylinder-head to fire the highly compressed gas. None of these methods were practical in their application to automotive engines because they did not permit flexible engine action which is so desirable. At the present time, electrical ignition systems in which the com-

pressed gas is exploded by the heating value of a small but intense electric arc or spark in the cylinder are standard, and the general practice seems to be toward the use of mechanical producers of electricity rather than chemical producers of current unless the batteries are used as an auxiliary to a generator and kept charged by it.

Electrical Ignition Best.—Two general forms of electrical ignition systems may be used, the most popular being that in which a current of electricity under high tension is made to leap a gap or air space between the points of one or more sparking plugs screwed into the combustion-chamber. The other form, which has been entirely abandoned in automobile and which was never used with airplane engine practice, but which is still used to some extent on large stationary or marine engines, is called the low-tension system because current of low voltage is used and the spark is produced by moving electrodes in the combustion-chamber, which come together and break to produce a spark. Because of this method of operation the system is known as a "make and break" ignition system but is obsolete in automotive engines and has never been used in airplane engines to the writer's knowledge.

The essential elements of any electrical ignition system, either high or low tension, are: First, a simple and practical method of current production; second, suitable timing apparatus to cause the spark in the combustion-chamber to occur at the right point in the cycle of engine action; third, suitable wiring and other apparatus to convey, control and intensify the current produced by the generator to the sparking member in the cylinder and fourth, a reliable and enduring sparking device in the combustion-chamber. The various appliances necessary to secure prompt ignition of the compressed gases should be described in some detail because of the importance of the ignition system. It is patent that the scope of a work of this character does not permit one to go fully into the theory and principles of operation of all appliances which may be used in connection with gasoline motor ignition, but at the same time it is important that the elementary principles be considered to some extent in order that the reader should have a proper understanding of the essential ignition apparatus. The first point considered will be the common methods of generating the electricity, then the appliances to utilize it and produce the required spark in the cylinder. Inasmuch as magneto ignition is almost universally used in connection with airplane engine ignition it will not be necessary to consider battery ignition system to the same extent, though it has advantages of merit and an outline of the Delco system and some typical diagrams will be given in proper sequence.

Fundamentals of Magnetism Outlined.—To properly understand the phenomena and forces involved in the generation of electrical energy by mechanical means it is necessary to become familiar with some of the elementary principles of magnetism and its relation to electricity. The following matter can be read with profit by those who are not familiar with the subject. Most persons know that magnetism exists in certain substances, but many are not able to grasp the terms used in describing the operation of various electrical devices because of not possessing a knowledge of the basic facts upon which the action of such apparatus is based.

Magnetism is a property possessed by certain substances and is manifested by the ability to attract and repel other materials susceptible to its effects. When this phenomena is manifested by a conductor or wire through which a current of electricity is flowing it is termed "electro-magnetism." Magnetism and electricity are closely related, each being capable of producing the other. Practically all of the phenomena manifested by materials which possess magnetic qualities naturally can be easily reproduced by passing a current of electricity through a body which, when not under electrical influence, is not a magnetic substance. Only certain substances show natural magnetic properties, these being iron, nickel, cobalt and their alloys, iron possessing it in the highest degree.

Magnetic Substances.—The earliest known substance possessing magnetic properties was a stone first found in Asia Minor. It was called the lodestone or leading stone, because of its tendency, if arranged so it could be moved freely, of pointing one particular portion toward the north. The compass of the ancient Chinese mariners was a piece of this material, now known to be iron ore, suspended by a light thread or floated on a cork in some liquid so one end would point toward the north magnetic pole of the earth. The reason that this stone was magnetic was hard to define for a time, until it was learned that the earth was one huge magnet and that the iron ore, being particularly susceptible to magnetism by induction, absorbed and retained some of this magnetism.

Magnetic Lines of Force.—Most of us are familiar with some of the properties of the magnet because of the extensive sale and use of small horseshoe magnets as toys. As they only cost a few pennies every one has owned one at some time or other and has experimented with various materials to see if they would be attracted. Small pieces of iron or steel were quickly attracted to the magnet and adhered to the pole pieces when brought within the zone of magnetic influence. It was soon learned that brass, copper, tin or zinc were not affected by the magnet. A simple experiment that serves to illustrate magnetic attraction of several substances is shown at A, Fig. 155. In this, several balls are hung from a standard or support, one of these being of iron, another of steel. When a magnet is brought near either of these they will be attracted toward it, while the others will remain indifferent to the magnetic force. Experimenters soon learned that of the common metals only iron or steel were magnetic.

If the ordinary bar or horseshoe magnet be carefully examined, one end will be found to be marked N. This indicates the north pole, while the other end is not usually marked and is the south pole. If the north pole of one magnet is brought near the south pole of another, a strong attraction will exist between them, this depending upon the size of the magnets used and the air gap separating the poles. If the south pole of one magnet is brought close to the end of the same polarity of the other there will be a pronounced repulsion or a lack of attraction of some force. These facts are easily proved by the simple experiment outlined at B, Fig. 155. A magnet will only attract or influence a substance having similar qualities. The like poles of magnets will repel each other because of the obvious impossibility of uniting two influences or forces of practically equal strength but flowing in opposite directions. The unlike poles of magnets attract each other be-

cause the force is flowing in the same direction. The flow of magnetism is through the magnet from south to north and the circuit is completed by the flow of magnetic influence through the air gap or metal armature bridging it from the north to the south pole.

Zone of Magnetic Influence Defined.—Magnets are commonly made in two forms, either in the shape of a bar or horseshoe. These two forms are made in two types, simple or compound. The latter are composed of a number of magnets of the same form united so the ends of like polarity are placed together, and such a construction will be more efficient and have more strength than a simple magnet of the same weight. The two common forms

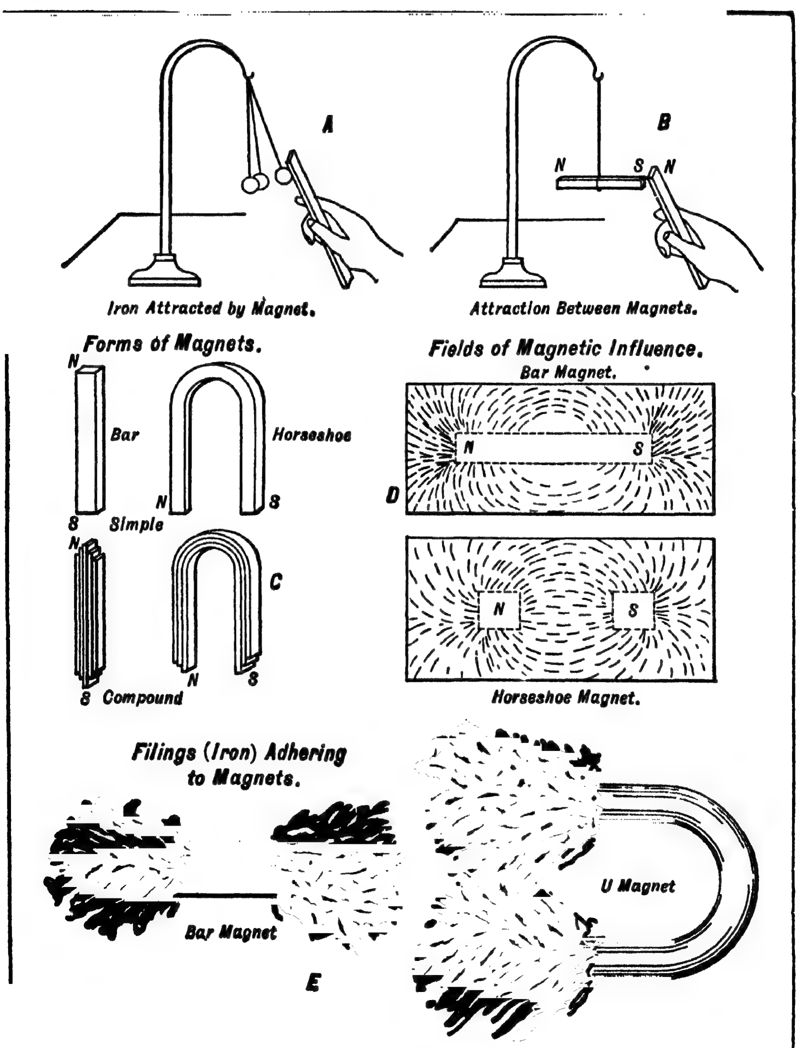


Fig. 155.—Two Simple Experiments to Demonstrate Various Magnetic Phenomena and to Clearly Outline Flow of Magnetism and Various Forms of Magnets.

of simple and compound magnets are shown at Fig. 155 C. The zone in which a magnetic influence occurs is called the magnetic field, and this force can be graphically shown by means of imaginary lines, which are termed "lines of force." As will be seen from the diagram at Fig. 155 D, the lines show the direction of action of the magnetic force and also show its strength, as they are closer together and more numerous when the intensity of the magnetic field is at its maximum. A simple method of demonstrating the presence of the force is to lay a piece of thin paper over the pole pieces of either a bar or horseshoe magnet and sprinkle fine iron filings on it. The particles of metal arrange themselves in very much the manner shown in the illustrations and prove that the magnetic field actually exists.

The form of magnet used will materially affect the size and area of the magnetic field. It will be noted that the field will be concentrated to a greater extent with the horseshoe form because of the proximity of the poles. It should be understood that these lines have no actual existence, but are imaginary and assumed to exist only to show the way the magnetic field is distributed. The magnetic influence is always greater at the poles than at the center, and that is why a horseshoe or U-form magnet is used in practically all magnetos. This greater attraction at the poles can be clearly demonstrated by sprinkling iron filings on bar and U magnets, as outlined at Fig. 155 E. A large mass gathers at the pole pieces, gradually tapering down toward the point where the attraction is least.

The Magnetic Circuit.—From the diagrams it will be seen that the flow of magnetism is from one pole to the other by means of curved paths between them. This circuit is completed by the magnetism flowing from one pole to the other through the magnet, and as this flow is continued as long as the body remains magnetic it constitutes a magnetic circuit. If this flow were temporarily interrupted by means of a conductor of electricity moving through the field there would be a current of electricity induced in the conductor every time it cut the lines of force. There are three kinds of magnetic circuits. A nonmagnetic circuit is one in which the magnetic influence completes its circuit through some substance not susceptible to the force. A closed magnetic circuit is one in which the influence completes its circuit through some magnetic material which bridges the gap between the poles. A compound circuit is that in which the magnetic influence passes through magnetic substances and nonmagnetic substances in order to complete its circuit.

How Iron and Steel are Made Magnetic.—Magnetism may be produced in two ways, by contact or induction. If a piece of steel is rubbed on a magnet it will be found a magnet when removed, having a north and south pole and all of the properties found in the energizing magnet. This is magnetizing by contact. A piece of steel will retain the magnetism imparted to it for a considerable length of time, and the influence that remains is known as residual magnetism. This property may be increased by alloying the steel with tungsten and hardening it before it is magnetized. Any material that will retain its magnetic influence after removal from the source of magnetism is known as a permanent magnet. If a piece of iron or steel is brought into the magnetic field of a powerful magnet it becomes a magnet without actual contact with the energizer. This is magnetizing

by magnetic induction. If a powerful electric current flows through an insulated conductor wound around a piece of iron or steel it will make a magnet of it. This is magnetizing by electro-magnetic induction. A magnet made in this manner is termed an electro-magnet and usually the metal is of such a nature that it will not retain but a small part of its magnetism when the current ceases to flow around it. Steel alloyed with tungsten is used in all cases where permanent magnets are required, while soft iron is employed in all cases where an intermittent magnetic action is desired. Magneto field magnets are always made of steel alloy, so treated that it will retain its magnetism for lengthy periods. Magneto magnets have been known to retain their magnetism, with a very slight loss, for periods of ten years when assembled on the magneto or stored with a keeper across the poles. A keeper is simply a bar or bridge of magnetic material.

Electricity and Magnetism Closely Related.—There are many points in which magnetism and electricity are alike. For instance, air is a medium that offers considerable resistance to the passage of both magnetic influence and electric energy, although it offers more resistance to the passage of the latter. Minerals like iron or steel are very easily influenced by magnetism and easily penetrated by it. When one of these is present in the magnetic circuit the magnetism will flow through the metal. Any metal is a good conductor for the passage of the electric current, but few metals are good conductors of magnetic energy. A body of the proper metal will become a magnet due to induction if placed in the magnetic field, having a south pole where the lines of force enter it and a north pole where they pass out.

We have seen that a magnet is constantly surrounded by a magnetic field and that an electrical conductor when carrying a current is also surrounded by a field of magnetic influence. Now if the conductor carrying a current of electricity will induce magnetism in a bar of iron or steel, by a reversal of this process, a magnetized iron or steel bar will produce a current of electricity in a conductor. It is upon this principle that the modern dynamo or magneto is constructed. If an electro-motive force is induced in a conductor by moving it across a field of magnetic influence, or by passing a magnetic field near a conductor, electricity is said to be generated by magneto-electric induction. All mechanical generators of the electric current using permanent steel magnets to produce a field of magnetic influence are of this type.

Basic Principles of Magneto Outlined.—The accompanying diagram, Fig. 156, will show these principles very clearly. As stated on a preceding page, if the lines of force in the magnetic field are cut by a suitable conductor an electrical impulse will be produced in that conductor. In this simple machine the lines of force exist between the poles of a horseshoe magnet. The conductor, which in this case is a loop of copper wire, is mounted upon a spindle in order that it may be rotated in the magnetic field to cut the lines of magnetic influence present between the pole pieces. Both of the ends of this loop are connected, one with the insulated drum shown upon the shaft, the other to the shaft. Two metal brushes are employed to collect the current and cause it to flow through the external circuit. It can be seen that when the shaft is turned in the direction of the

arrow the loop will cut through the lines of magnetic influence and a current will be generated therein. The pressure of the current and the amount produced vary in accordance to the rapidity with which the lines of magnetic influence are cut. The armature of a practical magneto, therefore, differs materially from that shown in the diagram. A large number of loops of wire would be mounted upon this shaft in order that the lines of magnetic influence would be cut a greater number of times in a given

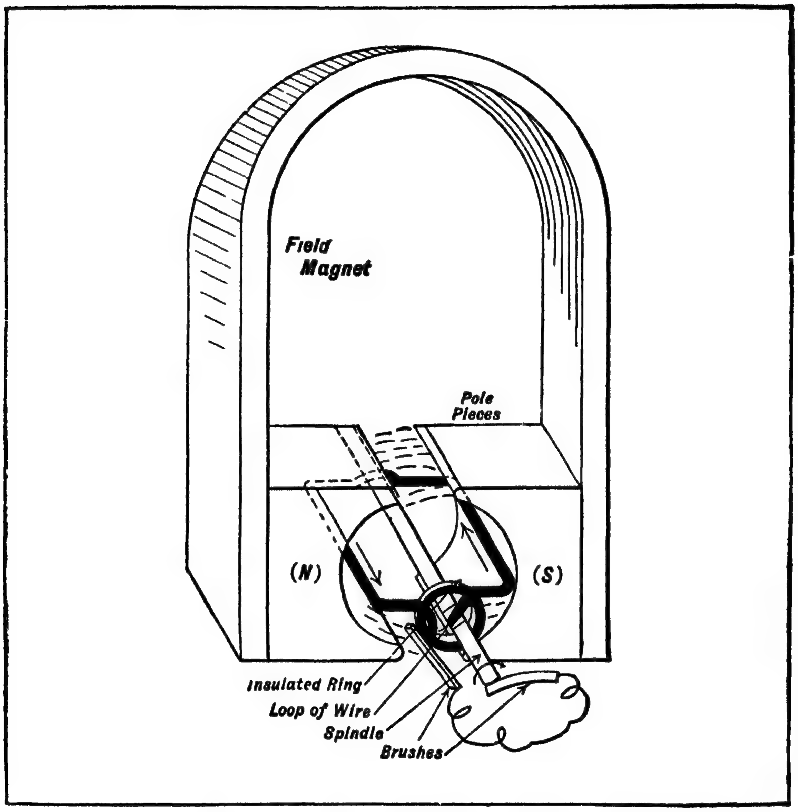


Fig. 156.—Arrangement of Magneto Showing Principal Parts, Simply to Make Method of Current Generation Clear.

period and a core of iron used as a backing for the wire. This would give a more rapid alternating current and a higher electro-motive force than would be the case with a smaller number of loops of wire.

The illustrations at Fig. 157 show a conventional double winding armature and field magnetic of a practical magneto in part section and will serve to more fully emphasize the points previously made. If the armature or spindle were removed from between the pole pieces there would exist a field of magnetic influence as shown at Fig. 155, but the introduction of this component provides a conductor (the iron core) for the magnetic energy, regardless of its position, though the facility with which the influence will be transmitted depends entirely upon the position of the core.

As shown at Fig. 157 A, the magnetic flow is through the main body in a straight line, while at B, which position the armature has attained after one-eighth revolution, or 45 degrees travel in the direction of the arrow, the magnetism must pass through in the manner indicated. At C, which position is attained every half revolution, the magnetic energy abandons the longer path through the body of the core for the shorter passage offered by the side pieces, and the field thrown out by the cross bar disappears. On further rotation of the armature, as at D, the body of the core again becomes energized as the magnetic influence resumes its flow through it.

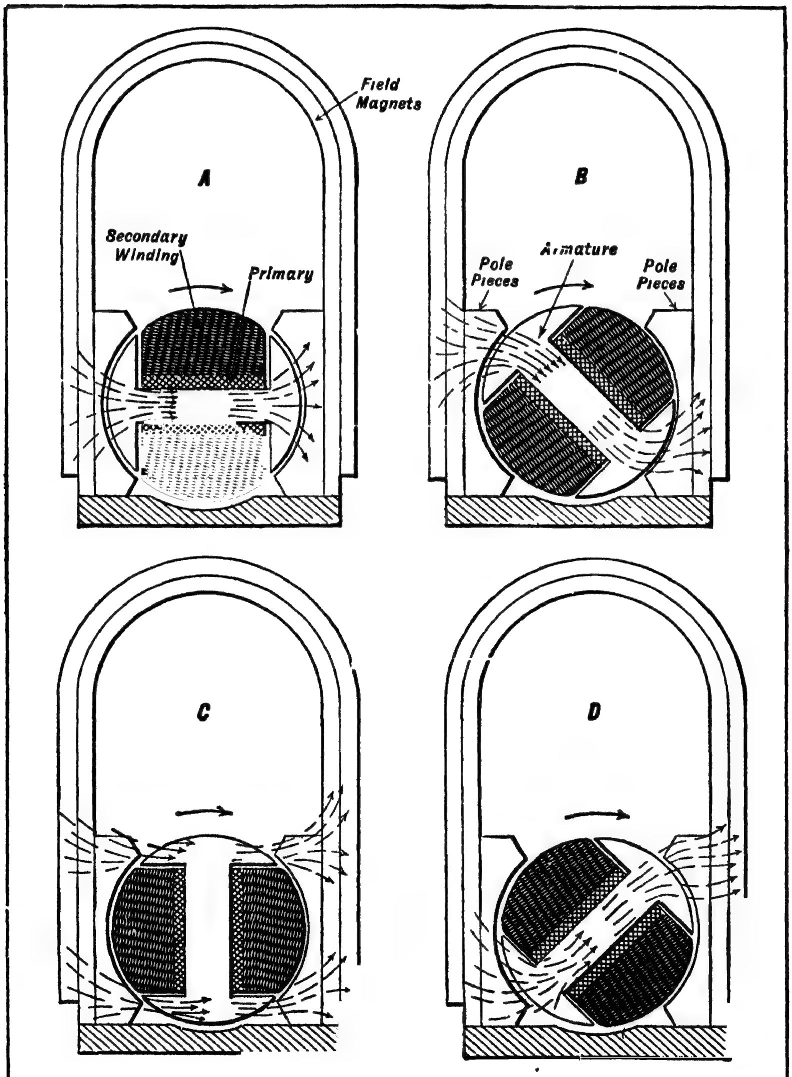


Fig. 157.—Showing How Strength of Magnetic Influence and the Current Induced in the Winding of a Shuttle Armature Vary with the Rapidity of the Changes of Flow.

These changes in the strength of the magnetic field when distorted by the armature core, as well as the intensity of the energy existing in the field, affect the windings, and the electrical energy induced therein corresponds in strength to the rapidity with which these changes in magnetic flow occur. The most pronounced changes in the strength of the field will occur as the armature passes from position B to D, because the magnetic field existing around the core will be destroyed and again re-established.

Why Magneto Must Be Timed.—During the most of the armature rotation the changes in strength will be slight and the currents induced in the wire correspondingly small; but at the instant the core becomes remagnetized, as the armature leaves position C, the current produced will be at its maximum, and it is necessary to so time the rotation of the armature that at this instant one of the cylinders is in properly charged condition to be fired. It is imperative that the armature be driven in such relation to the crankshaft that each production of maximum current coincides with the ignition point, this condition existing twice during each revolution of the armature, or at every 180 degrees travel. Each position shown corresponds to 45 degrees travel of the armature, or one-eighth of a turn, and it takes just three-eighths revolution to change the position from A to that shown at D. Thus, a four-cylinder engine using a two-pole or shuttle armature type magneto would require two sparks per revolution and the magneto armature would be driven at engine speed. The point of ignition in the combustion-chamber would coincide with that of maximum current generation in the magneto armature, driven by a positive drive in timed relation with the crankshaft. All magnetos do not use the wound shuttle or H armature. Some utilize a fixed winding as shown at Fig. 158 and a rotary inductor member made of two pieces of magnetic material. The core piece passing through the fixed winding has extending pole pieces and the rotary members complete the magnetic circuit at their various positions and produce rapid reversals of flow of magnetic energy through the core piece inside the winding as indicated by the arrows.

Essential Parts of a Shuttle Armature Magneto.—The magnets which produce the influence that in turn induces the electrical energy in the winding or loops of wire on the armature, and which may have any even number of opposed poles, are called field magnets. The loops of wire which are mounted upon a suitable drum and rotate in the field of magnetic influence in order to cut the lines of force is called an armature winding, while the core is the metal portion. The entire assembly is called the armature. The exposed ends of the magnets are called pole pieces and the arrangement used to collect the current is either a commutator or a collector. The stationary pieces which bear against the collector or commutator and act as terminals for the outside circuit are called brushes. These brushes are sometimes of copper, or some of its alloys, because copper has a greater electrical conductivity than any other metal. These brushes are nearly always of carbon, which is sometimes electroplated with copper to increase its electrical conductivity, though cylinders of copper wire gauze impregnated with graphite are utilized at times. Carbon is used because it is not so liable to cut the metal of the commutator as might be the case if the contact was of the metal to metal type. The reason for this is that carbon

has the peculiar property in that it materially assists in the lubrication of the commutator, and being of soft, unctuous composition, will wear and conform to any irregularities on the surface of the metal collector rings or commutator segments.

The magneto in common use consists of a number of horseshoe magnets which are compound in form and attached to suitable cast-iron pole pieces used to collect and concentrate the magnetic influence of the various magnets. Between these pole pieces an armature rotates. This is usually shaped like a shuttle, around which are wound coils of insulated wire. These are composed of a large number of turns and the current produced depends in great measure upon the size of the wire and the number of turns per coil. An armature winding of large wire will deliver a current of great amperage, but of small voltage. An armature wound with very fine wire will deliver a current of high voltage but of low amperage. In the ordinary form of magneto, such as used for ignition, the current is alternating in character and the break in the circuit should be timed to occur when the armature is at the point of its greatest potential or pressure. Where such a generator is designed for direct current production as for charging a storage battery the ends of the winding are attached to the segments of a commutator, but where the instrument is designed to deliver an alternating current one end of the winding is fastened to an insulator ring on one end of the armature shaft and the other end is grounded on the frame of the machine. The quantity of the current depends upon the strength of the magnetic field and the number of lines of magnetic influence acting through the armature. The electro-motive force varies as to the length of the armature winding, the number of coils or windings used and the number of revolutions at which the armature is rotated and cuts the lines of force. The current value in the inductor type magneto depends upon the relative number of turns of wire in the primary and secondary portions of the fixed coil and the strength and number of reversals of the magnetic flow through the core piece.

Transformer System Uses Low Voltage Magneto.—The magneto in the various systems which employ a transformer coil is very similar to a low-tension generator in general construction, and the current delivered at the terminals seldom exceeds 100 volts. As it requires many times that potential or pressure to leap the gap which exists between the points of the conventional sparkplug, a separate coil is placed in circuit to intensify the current to one of greater capacity. The essential parts of such a system and their relation to each other are shown in diagrammatic form at Fig. 158 A. As is true of other systems the magnetic influence is produced by permanent steel magnets clamped to the cast-iron pole pieces between which the armature rotates. At the point of greatest potential in the armature winding the current is broken by the contact breaker, which is actuated by a cam, and the current of higher value is induced in the secondary winding of the transformer coil when the low voltage current is passed through the primary winding. Most of the inductor type magnetos are transformer coil forms with the coil built in under the arch of the magnets. The magnetic flow is directly through the coil core, however, instead of passing through a wire wound armature of the shuttle type. It will be noted that

the points of the contact breaker are together except for the brief instant when separated by the action of the point of the cam upon the lever. It is obvious that the armature winding is short-circuited upon itself except when the contact points are separated. While the armature winding is thus short-circuited there will be practically no generation of current. When the points are separated there is a sudden flow of current through the primary winding of the transformer coil, inducing a secondary current in the other winding, which can be varied in strength by certain considerations in the preliminary design of the apparatus.

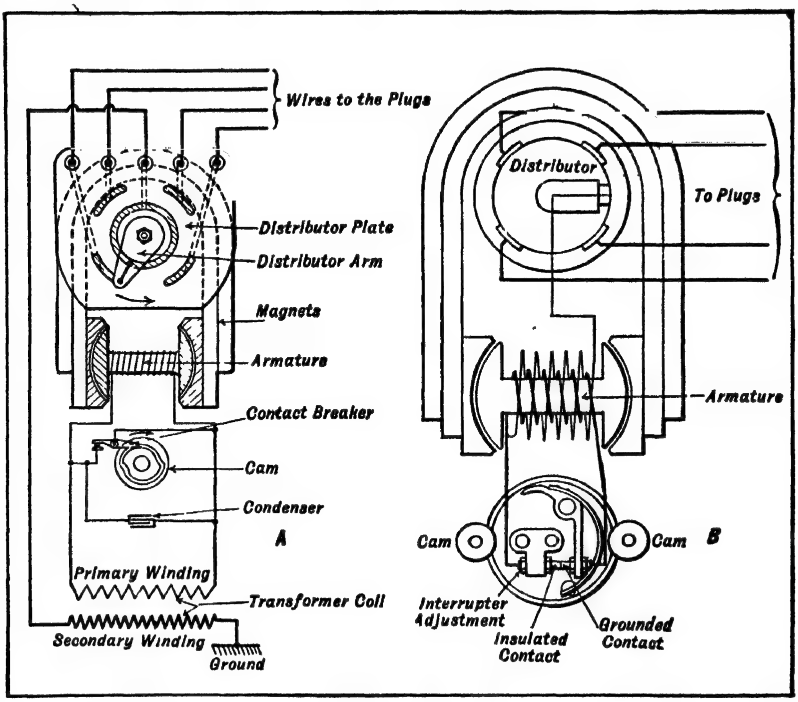


Fig. 158A.—Diagram Explaining Action of Low-Tension Magneto and Transformer Coil System. B—True High-Tension Magneto Ignition System.

Distribution of Secondary Current.—This current of higher potential or voltage is conducted directly to the plug if the device is fitted to a single-cylinder engine, or to the distributor rotor brush if fitted to a multiple-cylinder motor. The distributor consists of an insulator in which is placed a number of segments, one for each cylinder to be fired, and so spaced that the number of degrees between them correspond to the ignition points of the motor. A two-cylinder motor would have two segments, a three-cylinder, three segments, and so on within the capacity of the instrument. In the illustration a four-cylinder distributor is fitted, and the distributing arm is in contact with the segment corresponding to the cylinder about to be fired.

High-Tension Magnetos are Self-Contained.—The true high-tension magneto differs from the preceding inasmuch as the current of high voltage

is produced in the armature winding direct, without the use of the separate coil. Instead of but one winding, the armature carries two, one of comparatively few coils of coarse wire, the other of many turns of finer wire. The arrangement of these windings can be readily ascertained by reference to the diagram B, Fig. 158, which shows the principle of operation very clearly. The simplicity of the ignition system is evident by study of the diagram given at Fig 160. One end of the primary winding (coarse wire) is coupled or grounded to the armature core, and the other passes to the insulated part of the interrupter. While in some forms the interrupter or contact breaker mechanism does not revolve, the desired motion being imparted to the contact lever to separate the points of a revolving cam, in

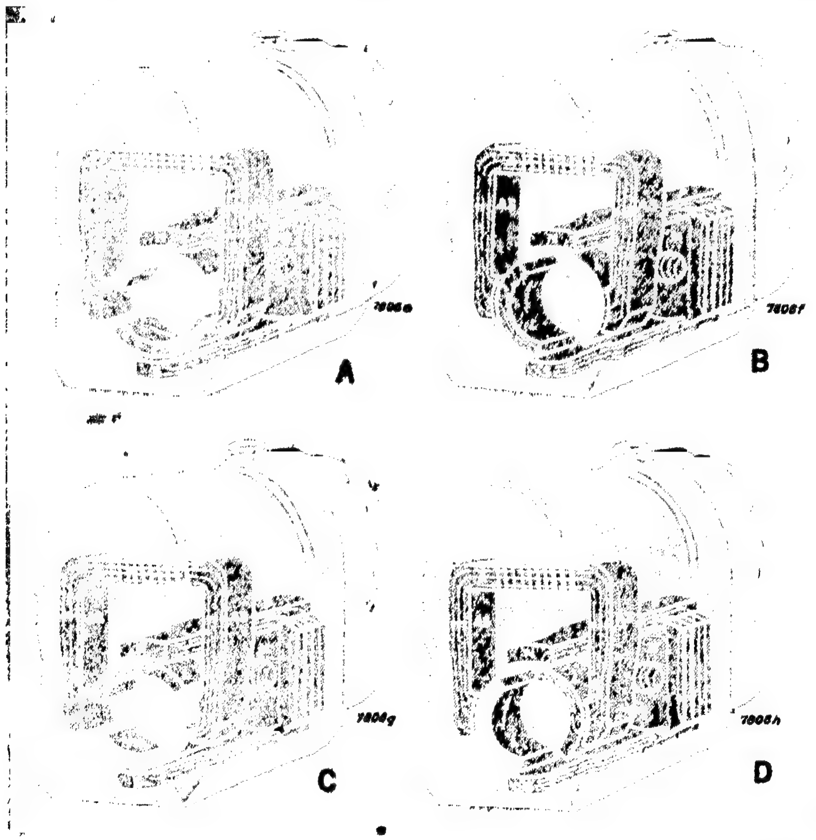


Fig. 158C.—How Rotary Inductor Member Changes Direction of Magnetic Flow Through a Fixed Winding.

this the cam or tripping mechanism is stationary or capable of only an oscillating or rocking motion to advance or retard the timing and the contact breaker revolves because it is joined to the armature shaft and must revolve with it and at the same speed This arrangement makes it possible to conduct the current from the revolving primary coil to the interrupter by a direct connection, eliminating the use of brushes, which would other-

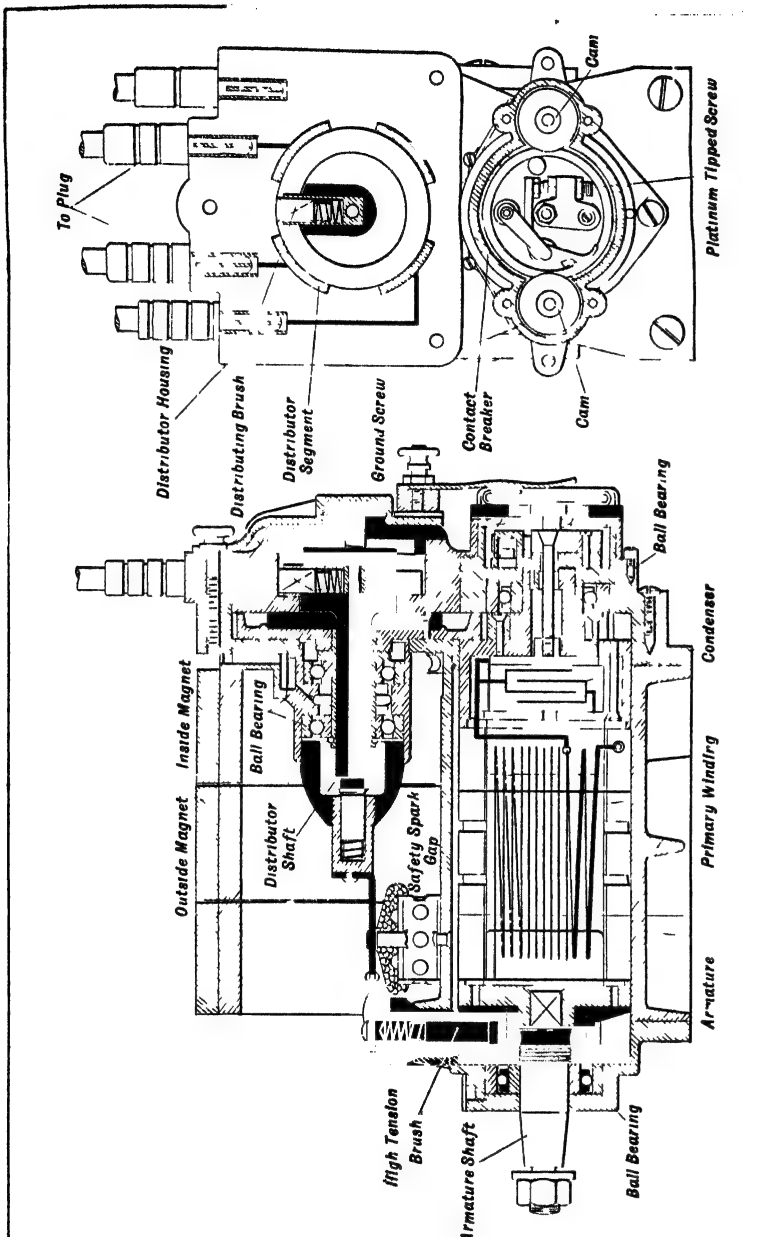


Fig. 159.—Side Sectional View of Early Bosch Automobile Type, High-Tension Magneto, Showing Disposition of Parts. End Elevation Depicts Arrangement of Distributor and Interruptor Mechanism.

wise be necessary. In other forms of this appliance where the winding is stationary, the interrupter may be operated by a revolving cam, though, if desired, the use of a brush at this point will permit this construction with a revolving winding.

Function of Make and Break.—During the revolution of the armature the grounded lever makes and breaks contact with the insulated point,

short-circuiting the primary winding upon itself until the armature reaches the proper position of maximum intensity of current production, at which time the circuit is broken, as in the former instance. One end of the secondary winding (fine wire) is grounded on the live end of the primary, the other end being attached to the revolving arm of the distributor mechanism as shown at Fig. 160. So long as a closed circuit is maintained feeble currents will pass through the primary winding, and so long as the contact points are together this condition will exist. When the current reaches its maximum value, because of the armature being in the best position, the cam operates the interrupter and the points are separated, breaking the short circuit which has existed in the primary winding.

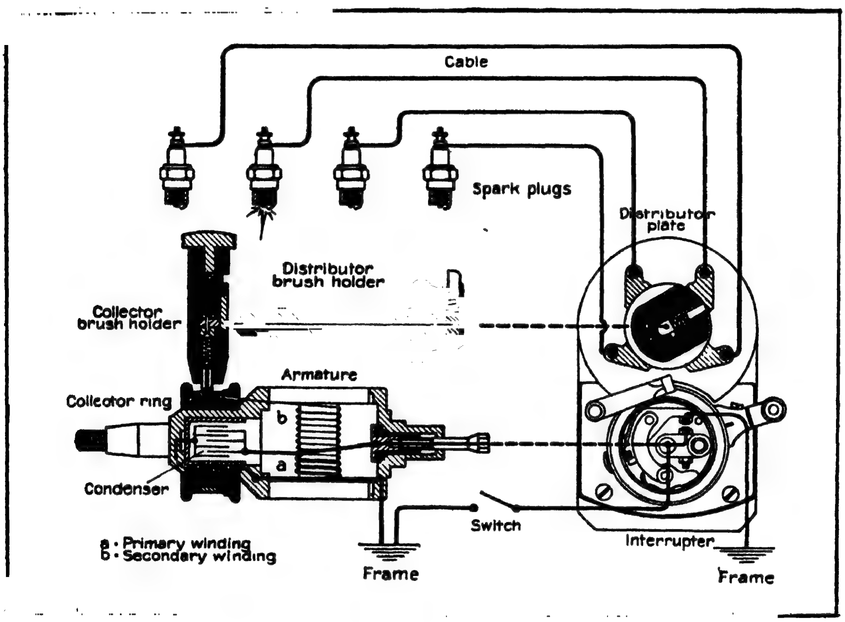


Fig. 160.—Simplified Diagram Showing Action of Robert Bosch High-Tension Magneto.

The secondary circuit has been open while the distributor arm has moved from one contact to another and there has been no flow of energy through this winding. While the electrical pressure will rise in this, even if the distributor arm contacted with one of the segments, there would be no spark at the plug until the contact points separated, because the current in the secondary winding would not be of sufficient strength. When the interrupter operates, however, the maximum primary current will be diverted from its short circuit and can flow to the ground only through the secondary winding and sparkplug circuit. The high pressure now existing in the secondary winding will be greatly increased by the sudden flow of primary current, and energy of high enough potential to successfully bridge the gap at the plug is thereby produced in the winding.

Advantages of Magneto Ignition.—The advantages of high tension magneto ignition may be summarized as follows:—

1. The magneto constitutes a self-contained spark generator, obtaining its primary current by magnetic induction.

2. Magneto ignition presents the minimum of fire hazard.

3. The spark produced by the aircraft magneto is well over the minimum energy required to ignite any fuel blend at any compression ratio and at any engine speed, consistent with modern aeronautical engine practice.

4. The primary and hence the secondary (high tension) current value increases in proportion to the speed of the magneto up to a certain point, where the value remains practically constant throughout the number of revolutions per minute required of the modern aeronautical engine.

5. Dependability, simplicity of construction, accessibility, ease of servicing, and low maintenance costs, are five of the features influencing the aeronautical engine designer in his choice of magnetos for ignition.

6. Practically every modern aeronautical engine is equipped with magnetos. In the great majority of cases two separate magnetos are mounted on each engine. There are two reasons for this: first, to provide two independent high-tension sparks, one to each of two sparkplugs located in each cylinder; second, to provide a double factor of safety.

Requirements of Aircraft Engine Ignition Systems.—The general requirements of aircraft ignition are reliability, low weight, compactness, low cost and adaptability. Since the flight of an airplane depends upon the continuous operation of the powerplant, reliability is undoubtedly the most important requirement of ignition equipment. The increased use of aircraft for commercial purposes is making the factor of cost more and more important, but, in aircraft, cost can never be placed ahead of reliability. This is not always the case in automotive vehicles. Light weight is also an important factor, since the weight of the ignition equipment is parasitic, that is, it does not add to the structural strength of the airplane. Compactness is important, for it usually means simplicity as well as reduction in weight. As a matter of production and maintenance cost, the question of adaptability of one model to different types of engine is of interest to the manufacturers both of ignition equipment and of engines. This subject was covered in a very complete manner by F. G. Shoemaker, associate mechanical engineer of the aircraft powerplant materiel division of the Air Corps, stationed at McCook Field, Dayton, Ohio, in a paper published in the July, 1927, *S. A. E. Journal* from which these extracts are taken.

Virtually all American aircraft engines with the exception of the Liberty-12, have been equipped with two single magnetos, most of which were supplied by the Scintilla Magneto Co., whose Type-AG magneto is regarded as the best single aircraft magneto available in production quantities in this country. The trend of development of all aircraft equipment is to reduce the size and weight and to increase the reliability and output; and this has been the object of the work on ignition done by the materiel division of the Air Corps.

Too great spark energy may cause "overlapping," which, with battery ignition, results in burning of the breaker contacts, and, in magneto ignition, reduces the intensity of alternate sparks. For ignition of supercharged engines at high altitude where the air density is much reduced, the air insulation of the ignition system is much less effective than at sea level, and

a flash-over distance to ground of roughly 0.75 in. is required. Coil failures will result unless the length of the coil is increased to provide this gap or all the air-spaces are filled with some insulating material.

Magneto vs. Battery Ignition.—The following brief statements outline the situation as regards the application of battery or magneto ignition to aircraft engines, no attempt being made to draw conclusions:

(1) No evidence indicates that the sparkplugs know where the sparks come from; therefore, one good spark is equal to another good spark
(2) Thousands of engines are operating satisfactorily on magneto and on battery ignition systems
(3) When electrical equipment is required for radio or lights, the battery ignition system should not be charged with any of the battery or generator weight, as this cannot be reduced by using magnetos. Therefore battery ignition is lighter than magneto ignition as now available
(4) When radio shielding is required, the additional shielding of the generator and of the lighting circuits, required with battery ignition, may offset the additional weight of magneto ignition
(5) The sparking rates of high-speed multiple-cylinder engines are approaching the limits obtainable from one present-type battery ignition-coil operating at from twelve to fifteen volts. The speed limitations of magneto ignition are principally mechanical and are not likely to be reached soon
(6) Battery ignition does not require a hand-starting magneto
(7) The sparking ability of a magneto is not dependent on a battery and therefore is not affected by long periods of storage or idleness
(8) A battery ignition system is cheaper than a magneto system.

This list can be added to indefinitely, but the above considerations are enough to show that the choice between battery and magneto ignition depends almost entirely on factors aside from that of spark intensity, which is the usual subject of controversy. The real factor to be considered is weight and if a storage battery and generator must be carried for lighting, starting or radio, then it is a moot question as to supplying magnetos or coil and distributor ignition. The following tabulation, given by Mr. Shoemaker gives comparative weights of various ignition systems applied to popular makes of aircraft engines.

COMPARATIVE WEIGHTS—BATTERY AND MAGNETO IGNITION:

Delco Ignition-Distributor on Curtiss D-12 Engine

2 Delco Ignition-Distributor Assemblies	10.0
2 Distributor-Drive Assemblies and Adapters	5.5
2 High-Tension Coils	4.4
1 Switch	1.2
Total	21.1

Splitdorf SS-12 Magneto on Curtiss D-12 Engine

2 Splitdorf SS-12 Magneto	28.2
2 Coupling and Drive Assemblies	3.0
2 Mounting-Brackets	2.5

1 Switch	1.2
1 Starting-Magneto	8.1
Total	43.0

Splitdorf VA-1 Double-Magneto on Curtiss D-12 Engine

1 Splitdorf VA-1 Magneto	15.5
1 Magneto Adapter-Flange and Drive-Coupling	1.9
2 Distributor-Drive Assemblies	4.5
2 Distributor-Heads and Rotors	3.0
1 Switch	1.2
1 Starting-Magneto	8.1
Total	34.2

Delco Ignition-Distributor for Packard 1500 and 2500 Engines

2 Ignition-Distributors	10.0
2 High-Tension Coils, Type D	4.4
1 Switch	1.0
Total	15.4

Scintilla AG 12-D Single Magnetos on Packard 1500 Engine

2 Scintilla-AG 12-D Single Magnetos	29.8
2 Coupling and Drive Assemblies	3.0
1 Mounting-Bracket	12.0
1 Switch	1.2
1 Starting-Magneto	8.1
Total	54.1

From the foregoing tabulation, it is seen that battery ignition, exclusive of generator and battery, weighs less than one-half that of the equivalent magneto ignition, when two single magnetos are used, and not more than two-thirds as much, when a double magneto is considered, though this does not seem to be an exactly fair comparison. With an ignition battery and generator, the battery ignition system has about the same weight as the double magneto installation. As the battery ignition system would be inoperative without a battery and as the battery requires constant charging by a generator to maintain its charge or current output strength, battery ignition is justified, in the opinion of the writer, only when a battery is needed for radio or electric starting and lighting systems as the total weight is as great as the double magneto installation. When an electric starter is used, the battery ignition method would be lighter because the one battery and generator would serve for ignition as well as starting.

Modern Engines Require Many Sparks.—The chief requirements of mechanical design are speed, rugged construction, simple mounting, light rotating parts, resistance to vibration, ample lubrication, freedom from moisture and fireproof ventilation. The first problem encountered in developing aircraft ignition apparatus is that of operating continuously at high speed. A twelve-cylinder aircraft engine cruising at 1,500 r.p.m. requires 9,000 sparks per minute, or 150 per second. In pursuit planes, when diving at full throttle, the engine speed may easily reach 3,000 r.p.m., requiring 18,000 sparks per minute, or 300 per second, whereas the normal speed for such engines is approximately 2,000 r.p.m. There is good reason

to believe that normal speeds of 3,000 r.p.m. will be reached in a few years so that the requirements for high speed are still rising.

In engines weighing two pounds per horsepower or less, the mass of the engine is small, compared with the inertia forces and torque reactions; consequently, vibrations of very high frequency are likely to be encountered. Such vibrations require only a short time in which to develop fatigue failures in parts such as mounting flanges, coil leads, condenser terminals, and the like, unless great care is taken to prevent repeated reversals of the

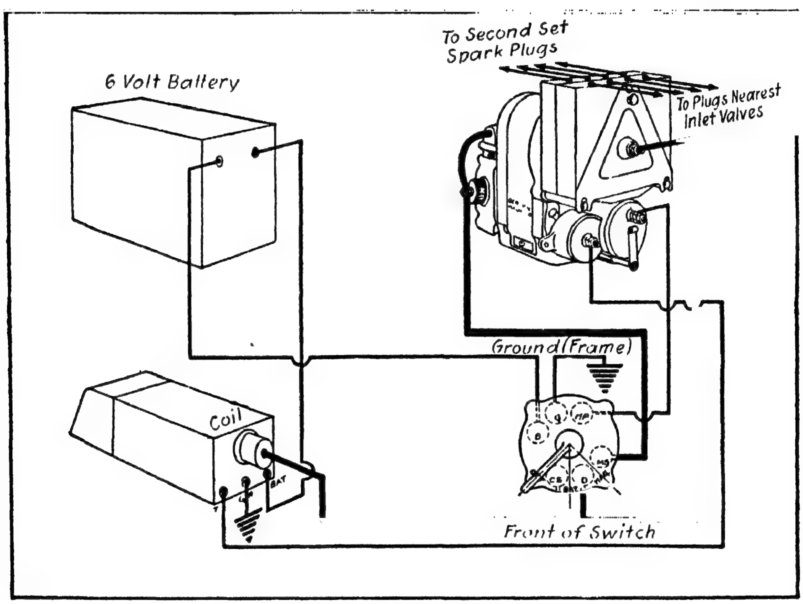


Fig. 161.—Berling Two Spark Ignition System.

stresses near the fatigue limit of the material. Experience has shown that ordinary automobile ignition equipment is entirely unsuited to aircraft service on this acount. The construction throughout must be of the most rugged type. Whenever possible, integral construction should be used, such as casting the body, pole-pieces and end frame of a magneto as a single unit. Studs with castellated nuts are preferable to screws and lock-washers for joining major parts. Apparently, every increase in engine speed produces a new list of vibration failures in equipment that had previously given no trouble whatever at the lower speeds.

Magneto Drive Important Problem.—As a result of the necessity for low weight, the crankshaft and the accessory drives are as small as possible and are therefore subject to rapid torsional vibrations. The twist of a twelve-cylinder engine crankshaft at full throttle may exceed ± two degrees at each power-impulse. Because of such variations in angular velocity transmitted to the ignition drive, it is essential that the moment of inertia of the rotating parts be kept as low as possible, to reduce the size and weight of the drive required. In some cases, it is necessary to introduce a

flexible coupling between the crankshaft and the magneto. This distortion is so serious with the Liberty-twelve engine that an attempt to use magnetos was abandoned on account of drive failures. A multitude of flexible magneto couplings are available but few are free from trouble in service.

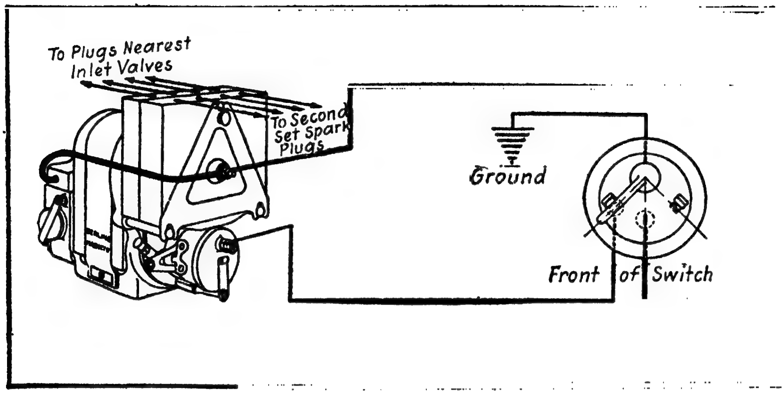


Fig. 162.—Berling Double Spark Independent System.

Too much flexibility allows the spark advance to wander about over a considerable range above and below the desired point. This may cause engine roughness, as the sparks that occur in the extreme advance position cause those particular cylinders to detonate. A very good drive for two magnetos is shown at Fig. 163, this being a foreign design and used on Lorraine aviation engines with great success.

If the flexible coupling can be dispensed with, a simple flange-mounting, with a driving-spline or gear directly on the rotor shaft, can be adopted.

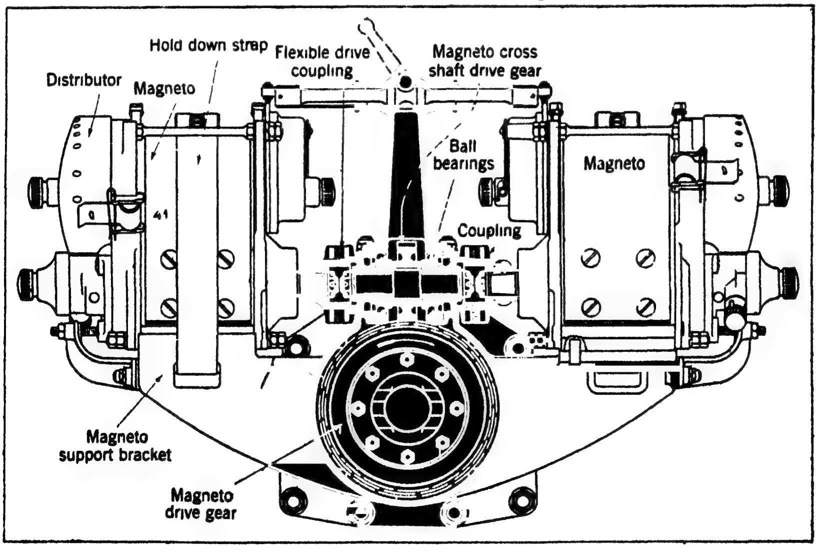


Fig. 163.—Typical Method of Driving Two Magnetos from a Cross Shaft Driven by Spiral Gearing from the Engine Crankshaft.

This reduces the necessity for a heavy shelf or bracket that is required with the conventional base-mounting, and the direct drive eliminates the use of an intermediate driveshaft assembly. By slotting the holes for the magneto flange-bolts, a simple means is provided for the accurate adjustment of the timing.

Lubrication Problem Difficult.—It is a very difficult problem to assure ample lubrication of the bearings, cam and breaker-arm while preventing oil from creeping into the distributor or on to the contacts. Both magnetos and distributors, as ordinarily installed, run at a fairly high temperature and the oil creeps wherever possible. The presence of parts carrying high voltage causes corona discharges, which, through their action on the oxygen and moisture in the air, cause the oil or grease to form gummy deposits on the bearings and exposed surfaces, unless the apparatus is well ventilated and the oil supply replenished. The presence of moisture inside a magneto or distributor is objectionable on account of the leakage of electricity across the moist insulation, and the rusting of the metal parts and bearing surfaces. The crankcase vapors are completely saturated with water vapor, which condenses as soon as it strikes a cool surface; and great care is necessary to prevent leakage of these vapors into the interior of the ignition system, particularly when a direct flange-mounting is used. Moisture may also form on the interior surface because of "breathing," or the passage of air into and out of the magneto as it cools off or becomes heated by the operation of the engine. To prevent this formation of moisture and to eliminate the corrosive gases, a certain amount of ventilation is required through properly located ventilating holes. But these ventilating holes are themselves not entirely innocent of trouble. They must be properly shielded to prevent flames from being blown out through the vent holes in case a leakage of gasoline occurs and the sparks in the distributor ignite the vapors in the magneto, thus causing an explosion.

Electrical Requirements Exacting.—The electrical requirements for aircraft ignition systems are considerably more exacting than those for other types of engine. The importance of engine reliability and the necessity for at least two sparkplugs in each cylinder, to assure rapid combustion without detonation, require two independent electrical sources of sparks. The high operating speeds of the engine require the magnetic and the electrical circuits to be designed for high-frequency operation. At these frequencies, the hysteresis and eddy-current losses in magnetos may cause an undue temperature rise in the coil and pole-pieces, unless the magnetic circuit is carefully laminated. Since many engines operate more smoothly when the two sparkplugs fire at slightly different degrees of advance, it is necessary to provide means for staggering the sparks. In some types of magneto, the eddy currents in the rotating parts and the high-frequency static discharges roughen the bearing surfaces and seriously shorten the life of the bearings. This necessitates insulating the ball-bearing races. It is generally thought that the greater the energy of the sparks, the better the ignition; and this is true for ordinary magneto or coil systems at moderate sparking rates. Some slight trouble may result from rapid burning of the electrodes of the sparkplugs when the sparks are very "fat" and the electrodes operate at a high temperature; but this in itself is not a sufficient

reason for limiting the spark energy. Before describing the more recent magneto designs, it may be well to review the features of earlier designs that have been used with engines that are still in use, such as the Curtiss OX series and contemporary powerplants.

The Berling Magneto.—The Berling magneto is a true high tension type delivering two impulses per revolution, but it is made in a variety of forms, both single and double spark. Its principle of action does not differ in essentials from the high tension type previously described. This magneto is used on Curtiss OX5 aviation engines and will deliver sparks in a positive

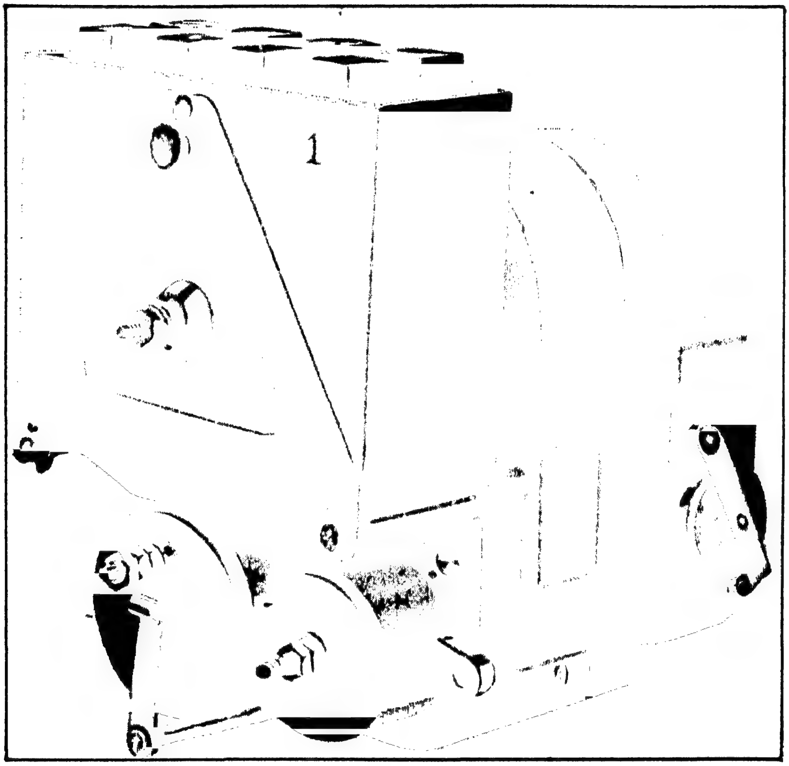


Fig. 164.—Type DD Berling High-Tension Magneto.

manner sufficient to insure ignition of engines up to 200 horsepower and at rotative speeds of the magneto armature up to 4,000 r.p.m. which is sufficient to take care of an eight-cylinder Vee-engine running up to 2,000 r.p.m. The magneto is driven at crankshaft speed on four-cylinder engines, at 1½ times crankshaft speed on six-cylinder engines and at twice crankshaft speed on eight-cylinder Vee-types. The types "D" and "DD" Berling magnetos are interchangeable with corresponding magnetos of other standard makes. The dimensions of the four-, six- and eight-cylinder types "D" and "DD" are all the same.

The ideal method of driving the magneto is by means of flexible direct connecting coupling to a shaft intended for the purpose of driving the magneto. As the magneto must be driven at a high speed, a coupling of some

flexibility is preferable. The employment of such a coupling will facilitate the mounting of the magneto, because a small inaccuracy in the lining up of the magneto with the driving shaft will be taken care of by the flexible coupling, whereas with a perfectly rigid coupling the line-up of the magneto must be absolutely accurate. Another advantage of the flexible coupling is that the vibration of the motor will not be as fully transmitted to the armature shaft on the magneto as in case a rigid coupling is used. This means prolonged life for the magneto.

The next best method of driving the magneto is by means of a gear keyed to the armature shaft. When this method of driving is employed, great care must be exercised in providing sufficient clearance between the

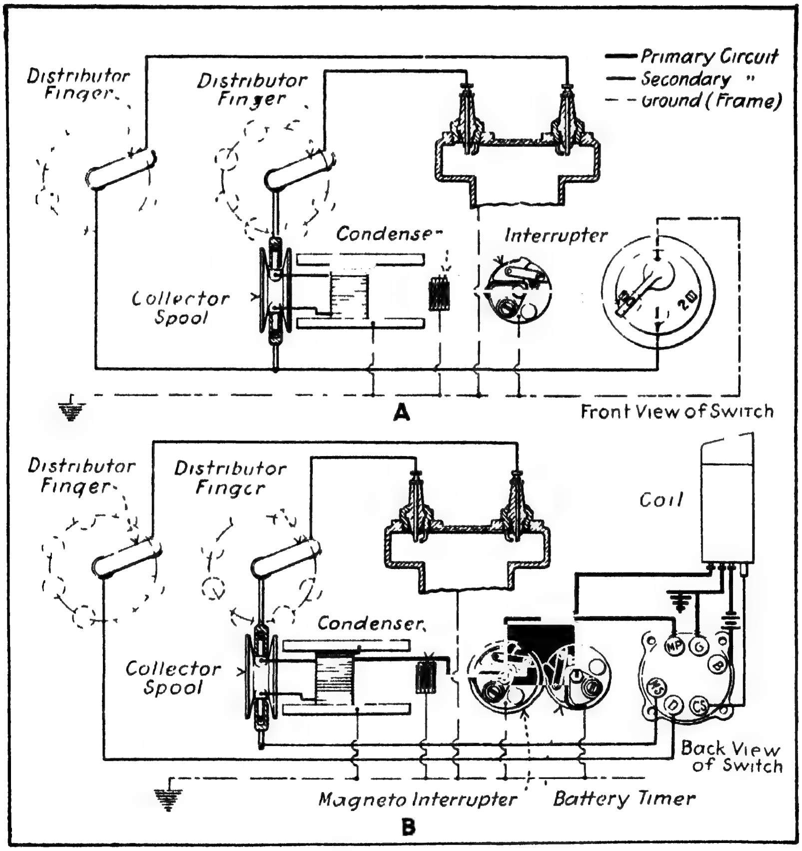


Fig. 165.—Wiring Diagram of Berling Magneto System.

gear on the magneto and the driving gear. If there should be a tight spot between these two gears it will react disadvantageously on the magneto. The third available method is to drive the magneto by means of a chain. This is the least desirable of the three methods and should be resorted to only in case of absolute necessity. It is difficult to provide sufficient clearance when using a chain without rendering the timing less accurate and positive.

Two Spark Independent Magneto.—Fig. 165 A, "A" shows diagrammatically the circuit of the "D" type two-spark independent magneto and the switch used with it. In position OFF the primary winding of the magneto is short-circuited and in this position the switch serves as an ordinary cut-out or grounding switch. In position "1" the switch connects the magneto in such a way that it operates as an ordinary single-spark magneto. In this position one end of the secondary winding is grounded to the body of the motor. This is the starting position. In this position of the switch the entire voltage generated in the magneto is concentrated at one sparkplug instead of being divided in half. With the motor turning over very slowly, as is the case in starting, the full voltage generated by the magneto will not in all cases be sufficient to bridge simultaneously two spark gaps, but is amply sufficient to bridge one. Also, this position of the switch tends to retard the ignition and should be used in starting to prevent backfiring. With the switch in position "2" the magneto applies ignition to both plugs in each cylinder simultaneously. This is the normal running position.

Two-Spark Dual Magneto.—Fig. 165 B shows diagrammatically the circuit of the type "DD" Berling high-tension two-spark dual magneto. This type is recommended for certain types of heavy duty airplane motors, which it is impossible to turn over fast enough to give the magneto sufficient speed to generate even a single spark of volume great enough to ignite the gas in the cylinder. The dual feature consists of the addition to the magneto of a battery interrupter. The equipment consists of the magneto, coil and special high-tension switch. The coil is intended to operate on six volts. Either a storage battery or dry cells may be used.

With the switch in the OFF position, the magneto is grounded, and the battery circuit is open. With the switch in the second or battery position marked "BAT," one end of the secondary winding of the magneto is grounded, and the magneto operates as a single-spark magneto delivering high-tension current to the inside distributor, and the battery circuit being closed the high-tension current from the coil is delivered to the outside distributor. In this position the battery current is supplied to one set of sparkplugs, no matter how slowly the motor is turned over, but as soon as the motor starts, the magneto supplies current as a single-spark magneto to the other set of the sparkplugs. After the engine is running, the switch should be thrown to the position marked "MAG." The battery and coil are then disconnected, and the magneto furnishes ignition to both plugs in each cylinder. This is the normal running position. Either a nonvibrating coil type "N-1" is furnished or a combined vibrating and nonvibrating coil type "VN-1."

Setting Berling Magneto.—The magneto may be set according to one of two different methods, the selection of which is, to some extent, governed by the characteristics of the engine, but largely due to the personal preference on the part of the user. In the first method described below, the most advantageous position of the piston for fully advanced ignition is determined in relation to the extreme advanced position of the magneto. In this case, the fully retarded ignition will not be a matter of selection, but the timing range of the magneto is wide enough to bring the fully retarded ignition after top-center position of the piston. The second method for the

setting of the magneto fixes the fully retarded position of the magneto in relation to that position of the piston where fully retarded ignition is desired. In this case, the extreme advance position of the magneto will not always correspond with the best position of the piston for fully advanced ignition, and the amount of advance the magneto should have to meet ideal requirements in this respect must be determined by experiment.

First Method:

1. Designate one cylinder as cylinder No. 1.

2. Turn the crankshaft until the piston in cylinder No. 1 is in the position where the fully advanced spark is desired to occur.

3. Remove the cover from the distributor block and turn the armature shaft in the direction of rotation of the magneto until the distributor finger-brush comes into such a position that this brush makes contact with the segment which is connected to the cable terminal marked "1." This is either one of the two bottom segments, depending upon the direction of rotation.

4. Place the cam housing in extreme advance, i.e., turn the cam housing until it stops, in the direction opposite to the direction of rotation of the armature. With the cam housing in this position, open the cover.

5. With the armature in the approximate position as described in "3," turn the armature slightly in either direction to such a point that the platinum points of the magneto interrupter will just begin to open at the end of the cam, adjacent to the fiber lever on the interrupter.

6. With this exact position of the armature, fix the magneto to the driving member of the engine.

Second Method:

1. Designate one cylinder as cylinder No. 1.

2. Turn the crankshaft until the piston in cylinder No. 1 is in the position at which the fully retarded spark is desired to occur.

3. Same as No. 3 under First Method.

4. Place the cam housing in extreme retard, i.e., turn the cam housing until it stops, in the same direction as the direction of rotation of the armature. With the cam housing in this position, open the cover.

5. Same as No. 5 under First Method.

6. Same as No. 6 under First Method.

Wiring the Magneto.—The wiring of the magneto is clearly shown by wiring diagram.

First determine the sequence of firing for the cylinders and then connect the cables to the sparkplug in the cylinders in proper sequence, beginning with cylinder No. 1 marked on the distributor block.

The switch used with the independent type must be mounted in such a manner that there will be a metallic connection between the frame of the magneto and the metal portion of the switch.

It is advisable to use a separate battery, either storage or dry cells, as a source of current for the dual equipment. Connecting to the same battery that is used with the generator and other electrical equipment may cause trouble, as a "ground" in this battery causes the coil to overheat.

Berling Magneto Lubrication.—Use only the very best of oil for the oil cups.

Put five drops of oil in the oil cup at the driving end of the magneto for every fifty hours of actual running.

Put five drops of oil in the oil cup at the interrupter end of the magneto, located at one side of the cam housing, for every hundred hours of actual running.

Lubricate the embossed cams in the cam housing with a thin film of vaseline every fifty hours of actual running. Wipe off all superfluous vaseline. Never use oil in the interrupter. Do not lubricate any other part of the interrupter.

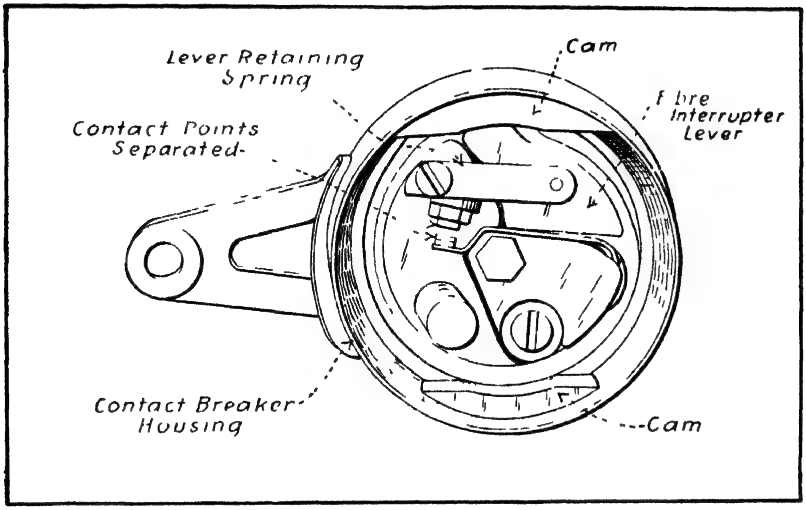


Fig. 166.—Berling Magneto Breaker Box Showing Contact Points Separated, and Interruptor Lever on Cam.

Adjusting the Interrupter.—With the fiber lever in the center of one of the embossed cams, as at Fig. 166, the opening between the platinum contacts should be not less than .016 inches and not more than .020 inches. The gauge riveted to the adjusting wrench should barely be able to pass between the contacts when fully open. The platinum contacts must be smoothed off with a very fine file. When in closed position, the platinum contacts should make contact with each other over their entire surfaces.

When inspecting the interrupter, make sure that the ground brush in the back of the interrupter base is making good contact with the surface on which it rubs.

Cleaning the Distributor.—The distributor block cover should be removed for inspection every 25 hours of actual running and the carbon deposit from the distributor finger-brush wiped off the distributor block by rubbing with a rag or piece of waste dipped in gasoline or kerosene. The high-tension terminal brush on the side of the magneto should also be carefully inspected for proper tension.

Locating Trouble.—Trouble in the ignition system is indicated by the motor "missing," stopping entirely, or by inability to start. It is safe to assume that the trouble is not in the magneto, and the carburetor, gasoline supply and sparkplugs should first be investigated.

If the magneto is suspected, the first thing to do is to determine if it will deliver a spark. To determine this, disconnect one of the high-tension leads from the sparkplug in one of the cylinders and place it so that there is approximately one-sixteenth inch between the terminal and the cylinder frame.

Remove the sparkplugs from the other cylinders to prevent the engine from firing and turn over the engine until the piston is approaching the end of the compression stroke in the cylinder from which the cable has been removed. Set the magneto in the advance position and rapidly rock the engine over the top-center position, observing closely if a spark occurs between the end of the high-tension cable and the frame.

If the magneto is of the dual type, the trouble may be either in the magneto or in the battery or coil system, therefore disconnect the battery and place the switch in the position marked "MAG." The magneto will then operate as an independent magneto and should spark in the proper manner. After this the battery system should be investigated. To test the operation of the battery and coil, examine all connections, making sure that they are clean and tight, and then with the switch in the "BAT," rock the piston slowly back and forth. If a type "VN-1" coil is used, a shower of sparks should jump between the high-tension cable terminal and the cylinder frame when the piston is in the correct position for firing. If no spark occurs, remove the cover from the coil and see that the vibrating tongue is free. If a type "N-1" coil is used, a single spark will occur. The battery should furnish six volts when connected to the coil, and this should also be verified. If the coil still refuses to give a spark and all connections are correct, the coil should be replaced and the defective coil returned to the manufacturer.

If both magneto and coil give a spark when tested as just described, the sparkplugs should be investigated. To do this, disconnect the cables and remove the sparkplugs. Then reconnect the cables to the plugs and place them so that the frame portions of the plugs are in metallic connection with the frame of the motor. Then turn over the motor, thus revolving the magneto armature, and see if a spark is produced at the spark gaps of the plugs.

The most common defects in sparkplugs are breaking down of the insulation, fouling due to carbon, or too large or small a spark gap. To clean the plugs a stiff brush and gasoline should be used. The spark gap should be about $\frac{1}{32}$ inch and never less than $\frac{1}{64}$ inch. Too small a gap may have been caused by beads of metal forming due to the heat of the spark. Too long a gap may have been caused by the points burning off. The sparkplug gap recommended by the engine builder should always be used because different engines have varying compression ratios and air gap resistance increases with augmented compression pressure.

If the magneto and sparkplugs are in good condition and the engine does not run satisfactorily, the setting should be verified according to instructions previously given, and, if necessary, readjusted.

Be careful to observe that both the type "VN-1" and type "N-1" coils are so arranged that the spark occurs on the opening of the contacts of the timer. As this is just the reverse of the usual operation, it should be carefully noted when any change in the setting of the timer is made. The timer on the dual type magneto is adjusted so that the battery spark occurs about five degrees later than the magneto spark. This provides an automatic advance as soon as the switch is thrown to the magneto position "MAG." This relative timing can be easily adjusted by removing the interrupter and shifting the cam in the direction desired.

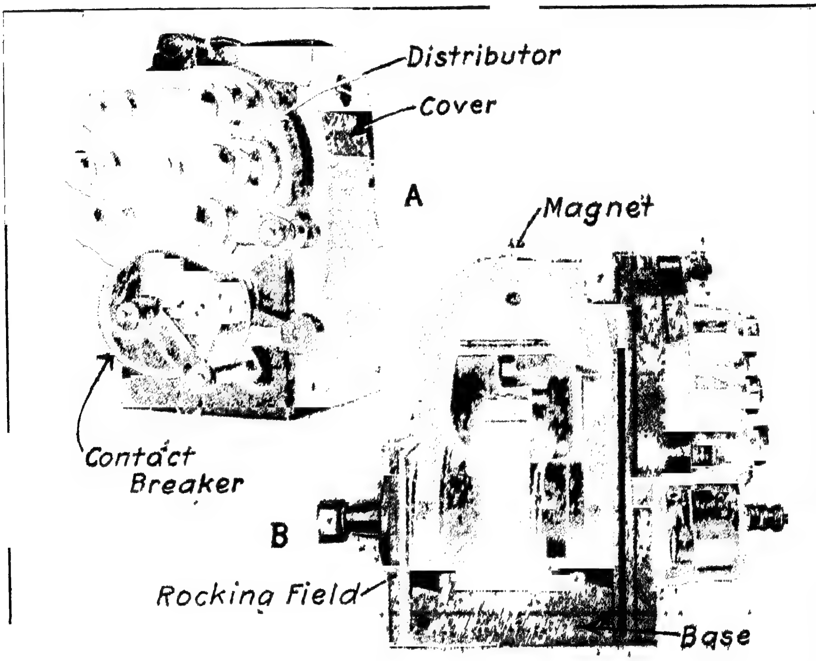


Fig. 167.—Dixie Model 60 Magneto for Six-Cylinder Airplane Engine Ignition.

The Dixie Magneto.—The Dixie magneto, shown at Fig. 167, operates on a different principle than the rotary armature type. It was used on the Hall-Scott and other aviation engines. In this magneto the rotating member consists of two pieces of magnetic material separated by a nonmagnetic center piece. This member constitutes true rotating poles for the magnet and rotates in a field structure, composed of two laminated field pieces, riveted between two nonmagnetic rings. The bearings for the rotating poles are mounted in steel plates, which lie against the poles of the magnets. When the magnet poles rotate, the magnetic lines of force from each magnet pole are carried directly to the field pieces and through the windings, without reversal through the mass of the rotating member and only a single air gap. There are no losses by flux reversal in the rotating part, such as take place in other machines, and this is said to account for the high efficiency of the instrument.

And this "Mason Principle" involved in the operation of the Dixie is simplified by a glance at the field structure, consisting of the nonmagnetic rings, assembled to which are the field pieces between which the rotating poles revolve (see Fig. 169). Rotating between the limbs of the magnets, these two pieces of magnetic material form true extensions to the poles of the magnets, and are, in consequence, *always* of the *same* polarity. It will be seen there is no reversal of the magnetism through them, and consequently no eddy current or hysteresis losses which are present in the usual rotor or inductor types. The simplicity features of construction stand out

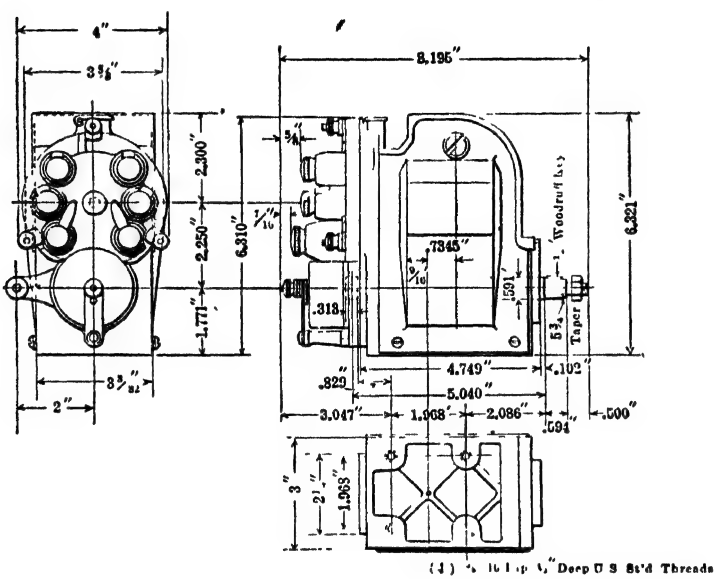


Fig. 168.—Installation Dimensions of Dixie Model 60 Magneto.

prominently here, in that there are no revolving windings, a detail entirely differing from the orthodox high-tension instrument. This simplicity becomes instantly apparent when it is found that the circuit breaker, instead of revolving as it does in other types, is stationary and that the whole breaker mechanism is exposed by simply turning the cover spring aside and removing cover. This makes inspection and adjustment particularly simple, and the fact that no special tool is necessary for adjustment of the platinum points—an ordinary small screwdriver is the whole "kit of tools" needed in the work of disassembling or assembling—is a feature of some value.

With dust- and water-protecting casing removed, and one of the magnets withdrawn, as in Fig. 170, the winding can be seen with its core resting on the field pole pieces and the primary lead attached to its side. An important feature of the high-tension winding is that the heads are of insulating material, and there is not the tendency for the high-tension current to jump to the side as in the ordinary armature type magneto. The high-

tension current is carried to the distributor by means of an insulated block with a spindle, at one end of which is a spring brush bearing directly on the winding, thus shortening the path of the high-tension current and eliminating the use of rubber spools and insulating parts. The moving parts of the magneto need never be disturbed if the high-tension winding is to be removed. This winding constitutes all of the magneto windings, no external spark coil being necessary The condenser is placed directly above the winding and is easily removable by taking out two screws, instead of being placed in an armature where it is inaccessible except to an expert, and where it cannot be replaced except at the factory whence it emanated.

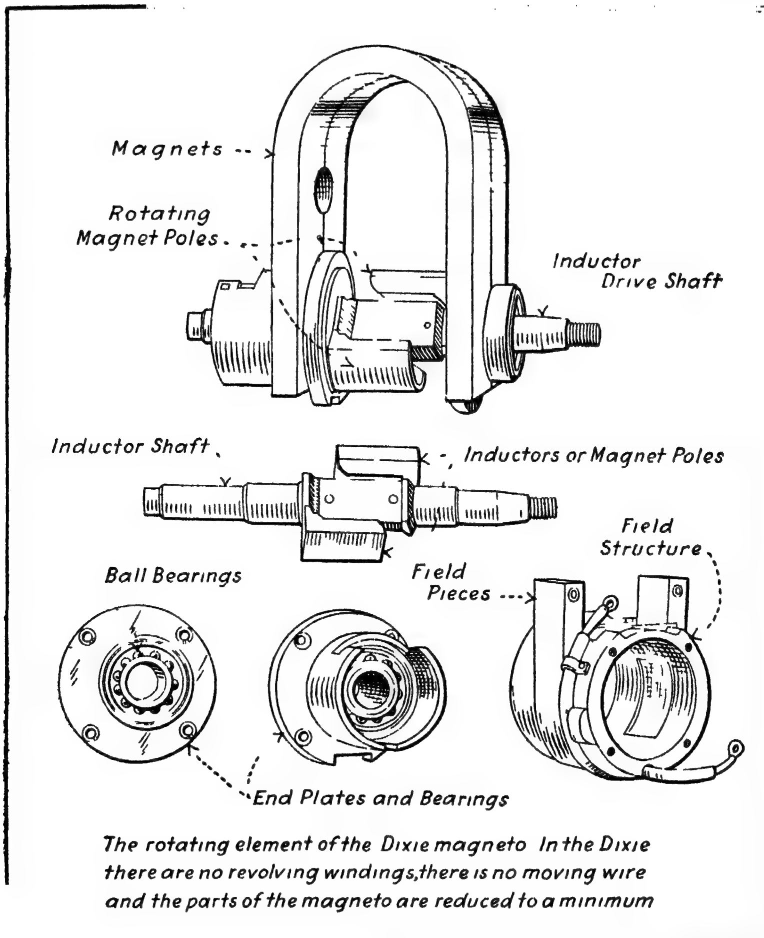


Fig. 169.—Rotating Elements of the Dixie Magneto.

Care of the Dixie Magneto.—The bearings of the magneto are provided with oil cups and a few drops of light oil every 1,000 miles are sufficient. The breaker lever should be lubricated every 1,000 miles with a drop of

light oil, applied with a toothpick. The proper distance between the platinum points when separated should not exceed .020 or one-fiftieth of an inch. A gauge of the proper size is attached to the screwdriver furnished with the magneto. The platinum contacts should be kept clean and properly adjusted. Should the contacts become pitted, a fine file should be used to smooth them in order to permit them to come into perfect contact. The distributor block should be removed occasionally and inspected for an ac-

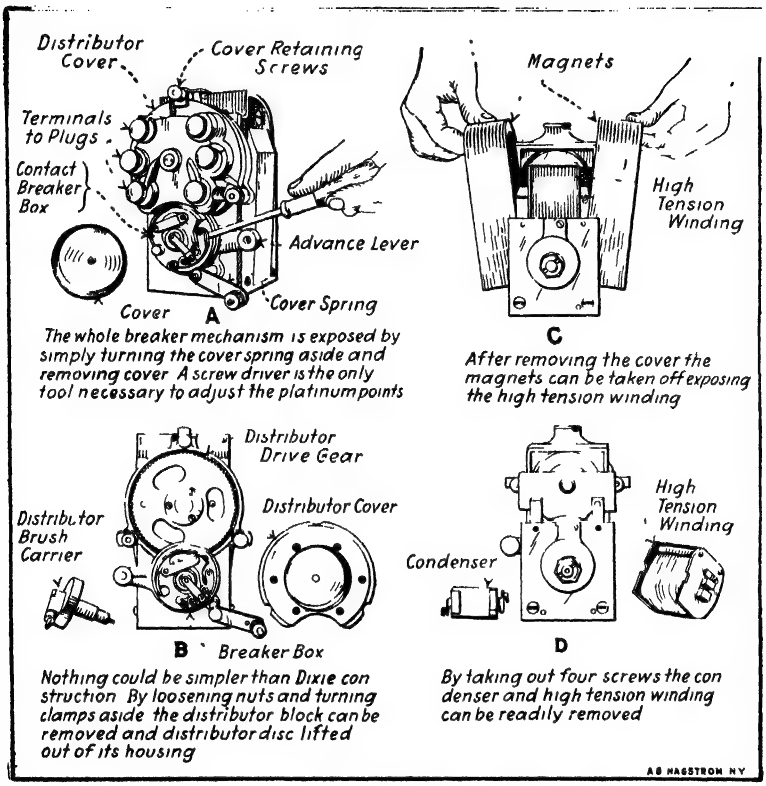


Fig. 170.—Suggestions for Adjusting and Dismantling Dixie Magneto. A—Screwdriver Adjusts Contact Points. B—Distributor Blocks Removed. C—Taking off the Magnets. D—Showing How Easily Condenser and High-Tension Wiring are Removed.

cumulation of carbon dust. The inside of the distributor block should be cleaned with a cloth moistened with gasoline and then wiped dry with a clean cloth. When replacing the block, care must be exercised in pushing the carbon brush into the socket. Do not pull out the carbon brushes in the distributor because you think there is not enough tension on the small brass springs. In order to obtain the most efficient results, the normal setting of the sparkplug points should not exceed 025 of an inch, and it is advisable to have the gap just right before a sparkplug is inserted.

The sparkplug electrodes may be easily set by means of the gauge attached to the screwdriver. *The setting of the sparkplug points is an important*

function which is usually overlooked, with the result that the magneto is blamed when it is not at fault.

Timing of the Dixie Magneto.—In order to obtain the utmost efficiency from the engine, the magneto must be correctly timed to it. This operation is usually performed when the magneto is fitted to the engine at the fac-

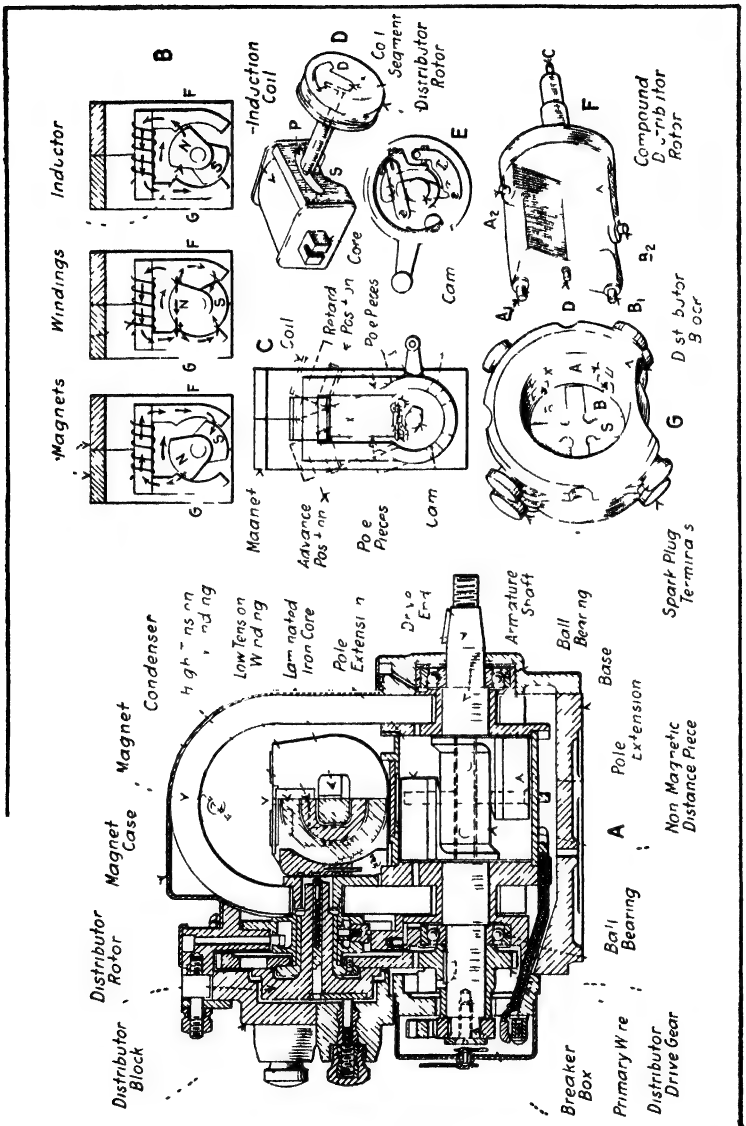


Fig. 171.—Sectional View Outlining Construction of Dixie Magneto with Compact Distribution for Eight-Cylinder Engine Ignition.

tory. The correct setting may vary according to individuality of the engine, and some engines may require an earlier setting in order to obtain the best results. However, should the occasion arise to retime the magneto, the procedure is as follows: Rotate the crankshaft of the engine until one of

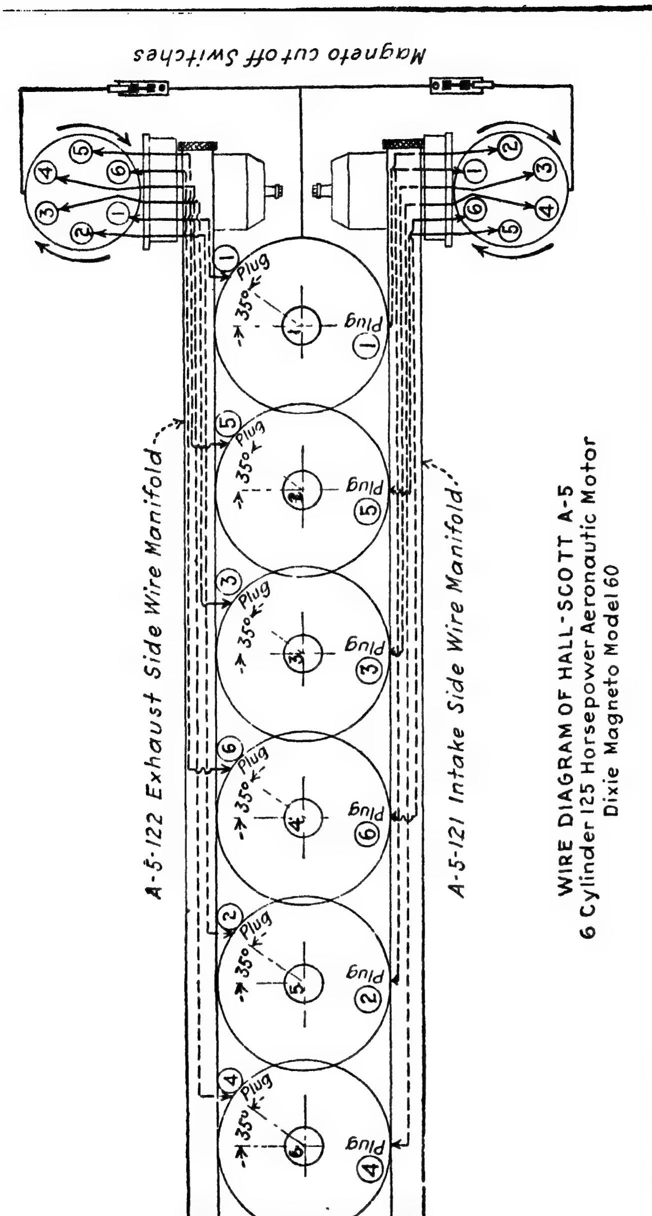


Fig. 172.—Wiring Diagram of Dixie Magneto Ignition of Six-Cylinder Aeronautic Motor.

the pistons, preferably that of cylinder No. 1, is one-sixteenth of an inch ahead of the end of the compression stroke. With the timing lever in full retard position, the driving shaft of the magneto should be rotated in the direction in which it will be driven. The circuit breaker should be closely observed and when the platinum contact points are about to separate, the drive gear or coupling should be secured to the drive shaft of the magneto.

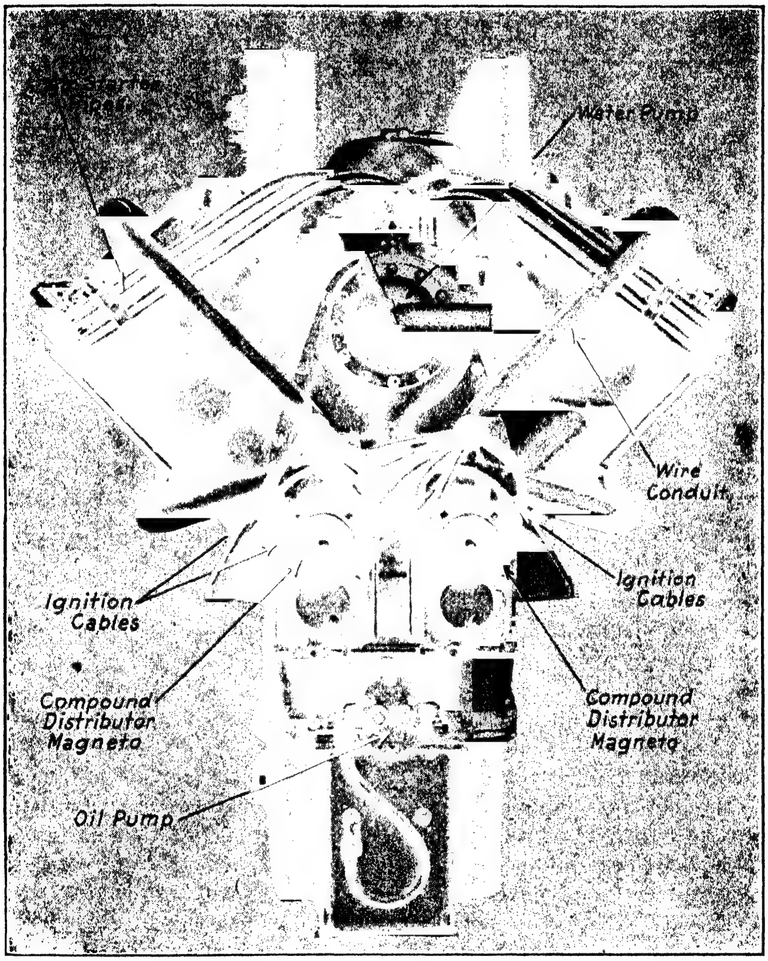


Fig. 173.—How Magneto Ignition was Installed on Early Thomas Morse 135 Horse-power "Vee" Engine.

Care should be taken not to alter the position of the magneto shaft when tightening the nut to secure the gear or coupling, after which the magneto should be secured to its base. Remove the distributor block and determine which terminal of the block is in contact with the carbon brush of the distributor finger and connect with plug wire leading to No. 1 cylinder to this terminal. Connect the remaining plug wires in turn according to the proper

sequence of firing of the cylinders. (See the wiring diagram for a typical six-cylinder engine at Fig. 172.) A terminal on the end of the cover spring of the magneto is provided for the purpose of connecting the wire leading to a ground switch for stopping the engine.

A special model or type of magneto is made for Vee-engines which use a compound distributor construction instead of the simple type on the model illustrated and a different interior arrangement permits the production of four sparks per revolution of the rotors. This makes

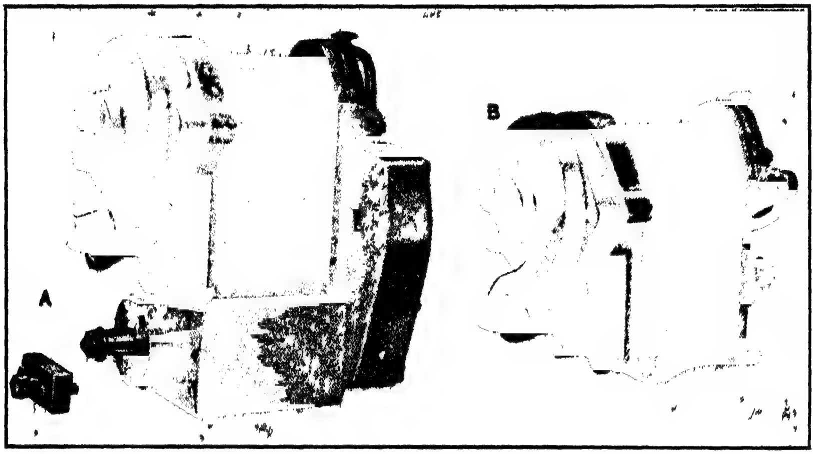


Fig. 174.—Robert Bosch Magneto for Use with Curtiss OX Series Engines.

it possible to run the magneto slower than would be possible with the two-spark form The application of two compound distributor magnetos of this type to an early Thomas-Morse 135 horsepower motor of the eight-cylinder Vee pattern is clearly shown at Fig 173 This type of engine is obsolete and the Splitdorf magneto has been changed in the latest forms as will be described later in another chapter.

Robert Bosch Magnetos.—The Curtiss OX series engines were provided with a variety of magnetos, two types of Robert Bosch magneto installations being shown at Fig. 174. The Bosch HL 8bRS52 shown at A generated four sparks per revolution. As the magneto drive of the Curtiss OX5 engine runs at twice crankshaft speed, it is necessary to provide a gear reduction between the original magneto drive and the Bosch magneto HL8b. An auxiliary shaft, supported by two ball bearings in an intermediate bracket, drives the reduction gears which are enclosed in a gear housing. For noiseless operation and to prevent rapid wear, the large gear is made of micarta. Since the gear housing is filled with lubricating grease, the gears do not require attention for a long time. The ball bearings of the auxiliary shaft need no attention at all because they are packed with a special heat resisting grease US 501, which must be replaced only when the unit is disassembled for a general overhauling. The magneto can be furnished with a Hand Starting Magneto Type AM I/1 and Switch WZ 14016, both of which can be mounted on the instrument board near the pilot's seat.

The Curtiss OXX6 engine requires two of the HL 8bS50 magnetos shown at Fig. 174 B. The one in front rotates anti-clockwise, the rear magneto rotates clockwise. They are provided with a special base plate.

The magnetos can be furnished with a Hand Starting Magneto and Switch, both of which can be installed as previously indicated. The installation of the hand starting magneto provides easy starting, even if the engine is cold and when there is no help available to throw over the propeller. As soon as the engine cylinders contain ignitable mixture, a few turns on

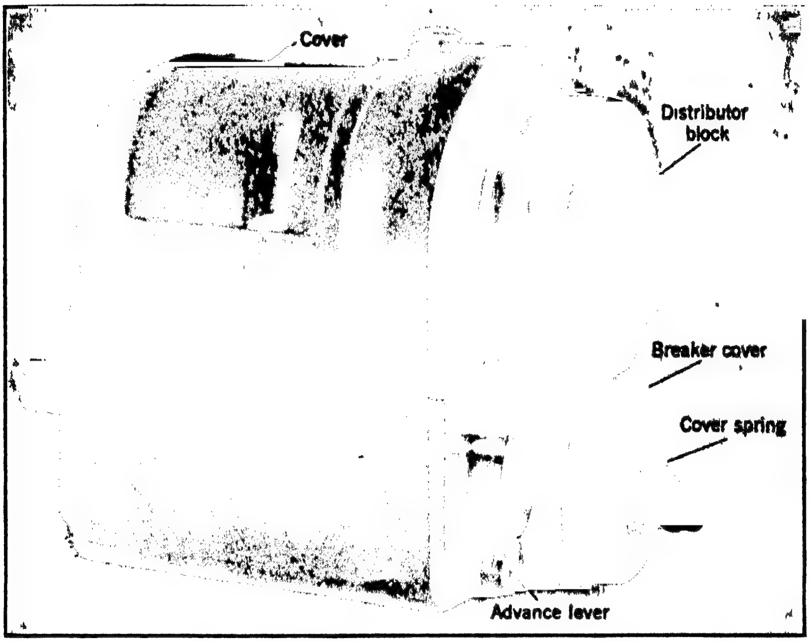


Fig. 175.—Robert Bosch Type GF-9 Magneto for Radial Aircraft Engines.

the crank of the hand starting magneto suffice to put the engine into operation. Auxiliary distributor electrodes of the HL 8 magnetos distribute the high-tension starting current to those cylinders which, in their combustion stroke, have passed top dead center. Backfiring is thus avoided when starting.

Realizing the need for a magneto suited for radial cylinder aircraft engines, the Type GF Super Energy magneto shown at Figs. 175 and 176 has been developed. This uses a rotary inductor member instead of a wound armature. The arrangement of parts is clearly shown at Fig. 176 A, which is a phantom view and the ease of disassembly is shown at B, where the various parts comprising the assembly at A are separated but placed in the relation they occupy relative to each other when assembled. The action can be readily understood by reviewing elementary explanatory matter previously presented.

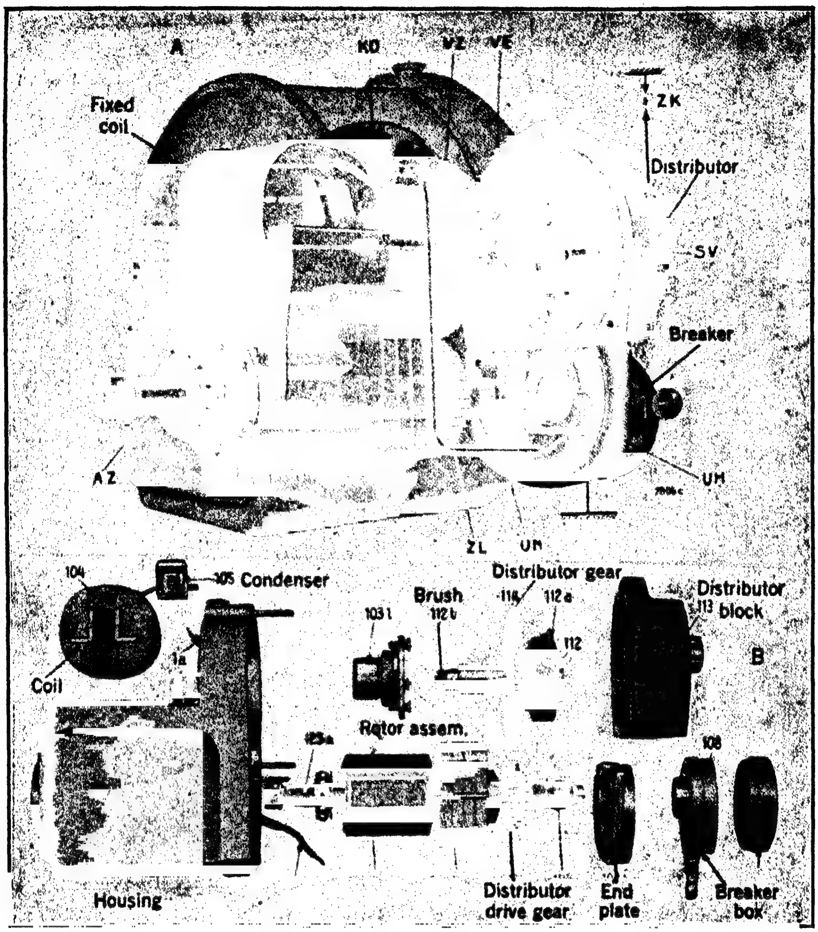


Fig. 176.—Illustration Showing Construction of Robert Bosch Type GF Magneto.

QUESTIONS FOR REVIEW

1. Outline possible methods of gasoline engine ignition.
2. Why is electric ignition best?
3. What are the common types of electrical ignition?
4. Name advantages of magneto ignition.
5. Name advantages of battery ignition.
6. Outline action of true high-tension magneto.
7. How and why is a high-tension magneto timed?
8. Do all magnetos use rotary windings?
9. How does inductor type magneto work?
10. When is battery ignition used for airplane work?

CHAPTER XIV

SCINTILLA AIRCRAFT MAGNETOS

Characteristics of Scintilla Design—Parts of Scintilla Magneto—The Rotating Magnet —The Contact Breaker Assembly—The Front End Plate—The Coil—The Magneto Housing—The Main Cover—The Distributor Blocks—The Breaker Cover—Electrical Operation of Magnetos—High Tension Current—Safety Gap—Booster Connections for Starting—Stopping the Engine—Taking Down Scintilla Magneto—Cleaning Scintilla Parts—Inspection of Scintilla Magneto—Front End Plate—Breaker Assembly—Rotating Magnet Assembly—Assembly of Scintilla Magneto —Testing Magneto After Assembly—Electrical Tests—Charging Magneto—Installing Scintilla Magneto—Timing Magneto—Changing Direction of Rotation—Adjusting End Play of Rotating Magnet—Installing Outer Bearing Races—Contact Points—Adjusting Distributor Gear—Timing Magneto by Lights—Oiling the Magneto—Shipment and Storage—Type V-AG Magneto—Inspection and Assembly of Distributor Gear Ball Bearing Assembly—Scintilla Magneto Types—Use of Scintilla Magneto Tool Set.

Characteristics of Scintilla Design.—The principle of design and construction of the Scintilla Aircraft Magneto is such that its chief characteristics lie in the complete inversion of the systems hitherto used. The simplicity of this design and the numerous advantages to be had in a magneto of this type are obvious. A cut away section is shown at Fig. 178 showing the various parts in their correct relation and duplicates of these parts placed about the magneto so their construction can be seen. The view at Fig. 177 permits of more detailed study than the smaller view at Fig. 178. The parts are also shown in the sectional view at Fig. 179 for the benefit of those of our readers capable of reading engineering drawings.

Instead of the horseshoe magnets and inductor commonly employed, the Scintilla Aircraft Magneto uses one rotating magnet. It is made of special chrome magnet steel and has either two or four poles, depending upon the type magneto in which it is used. The pole extremities are laminated, the laminations being held in place by an end plate on which the breaker cam is mounted. The rotation of the magnet produces reversals of magnetic flux through the core of the coil. Due to the close proximity of the poles to each other this type magnet retains its magnetic strength for long periods of time. The drive shaft is turned out of high-tensile strength steel pressed into the magnet and secured by a pin through the magnet and shaft. This construction assures the maximum toughness and ability to withstand the severe driving stresses imposed upon it.

The contact breaker mechanism is of the rocker lever type actuated by either a two or four lobe cam. The cage on which the various parts are assembled to form the complete breaker mechanism can be readily removed from the magneto without the use of any tools whatever. To eliminate the external connections necessary for the conventional condenser application and to provide adequate protection for this most important part of the ignition device, the condenser has been incorporated within the coil and located between the primary and secondary windings.

All primary connections within the magneto are made by means of laminated leaf springs thus insuring positive and reliable circuit connec-

tions. The internal electric and magneto circuits are shown in sectional view at Fig. 180. All Scintilla Aircraft Magnetos, the type designation of which ends with a "D," have provision made for booster starting. This connection permits the introduction of high-tension current from an external source, thus facilitating the starting of the engine. The contact points are of the finest material that it is possible to obtain, the alloy being 75 per cent platinum and 25 per cent iridium.

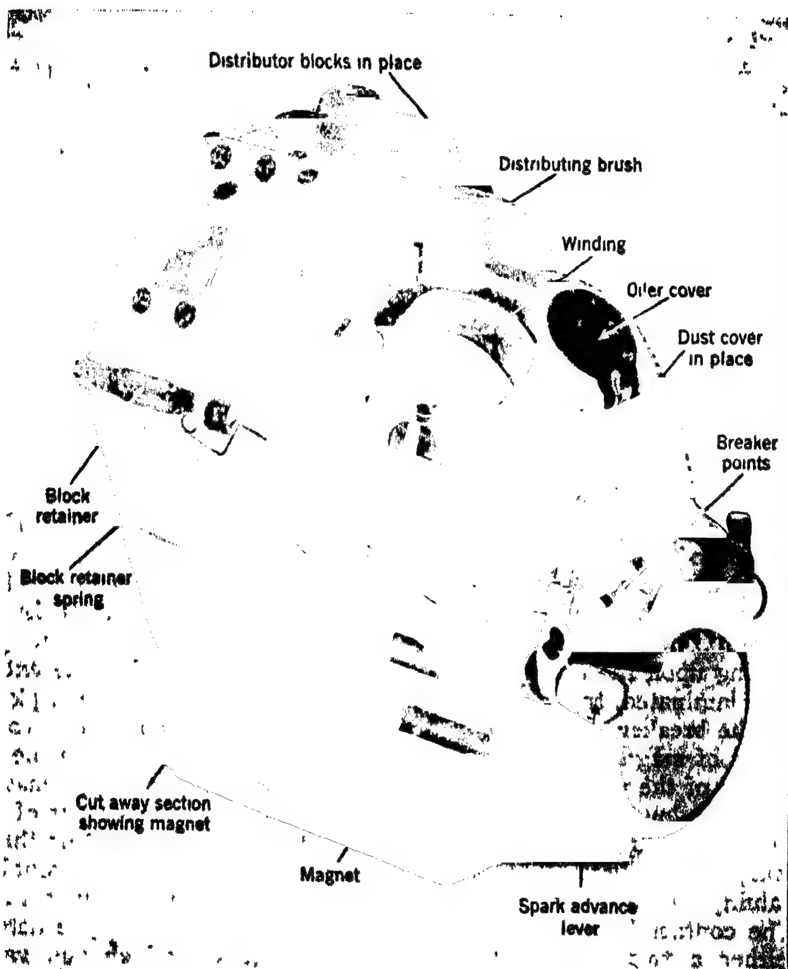


Fig. 177.—Cut-Away View of Scintilla Aircraft Magneto Showing Rotating Magneto Assembly, Interruptor Points, and Location of Fixed Coils.

Parts of Scintilla Magneto.—The Scintilla Aircraft Magneto, when disassembled for inspection or repair, will consist of the following subassemblies:

1. The Rotating Magnet
2. The Contact Breaker

4. The Coil
5. The Magneto Housing
6. The Main Cover with Booster and Ground Connection Block
7. The Distributor Blocks
8. The Breaker Cover

A view of the magneto partially dismantled is given at Fig. 181 which shows some of the parts mentioned removed from the assembly. The arrangement of these parts on the finished magneto can be studied by careful examination of the diagrams at Figs. 179 and 180.

The Rotating Magnet.—The magnet is supported in the magneto housing by the drive end bearing and the breaker end bearing. End play

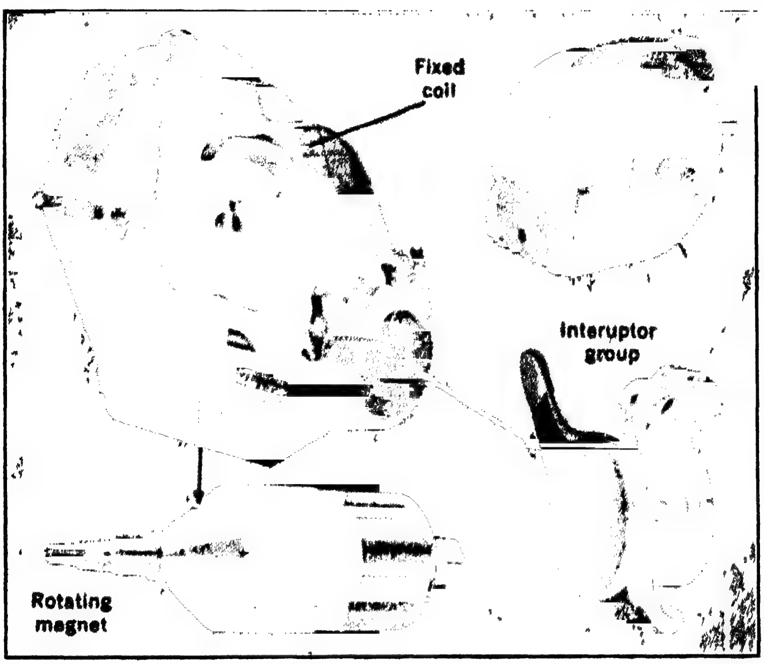


Fig. 178.—Scintilla Aircraft Magneto Parts and Location in Assembly.

is adjusted by spacing washers behind each inner ball race. The drive end shaft carries the inner race for the drive end bearing and the small distributor gear which is keyed to the drive end shaft. The breaker end shaft carries the inner race for the breaker end bearing and the breaker cam. The breaker cam is keyed on a taper shaft and secured by a screw.

The Contact Breaker Assembly.—This mechanism is carried by the breaker cage. The breaker cage has its ground and compensating springs riveted to it. It carries the breaker lever and its axle and the short contact screw. The insulated support carrying the long contact screw and fiber stop is mounted at the top of the breaker cage. The flat spiral bayonet lock spring lies between the back of the breaker cage and the dog plate and inside the bayonet lock latch. The oil wick for the cam is located in the bottom of the breaker cage. The main spring for the contact breaker

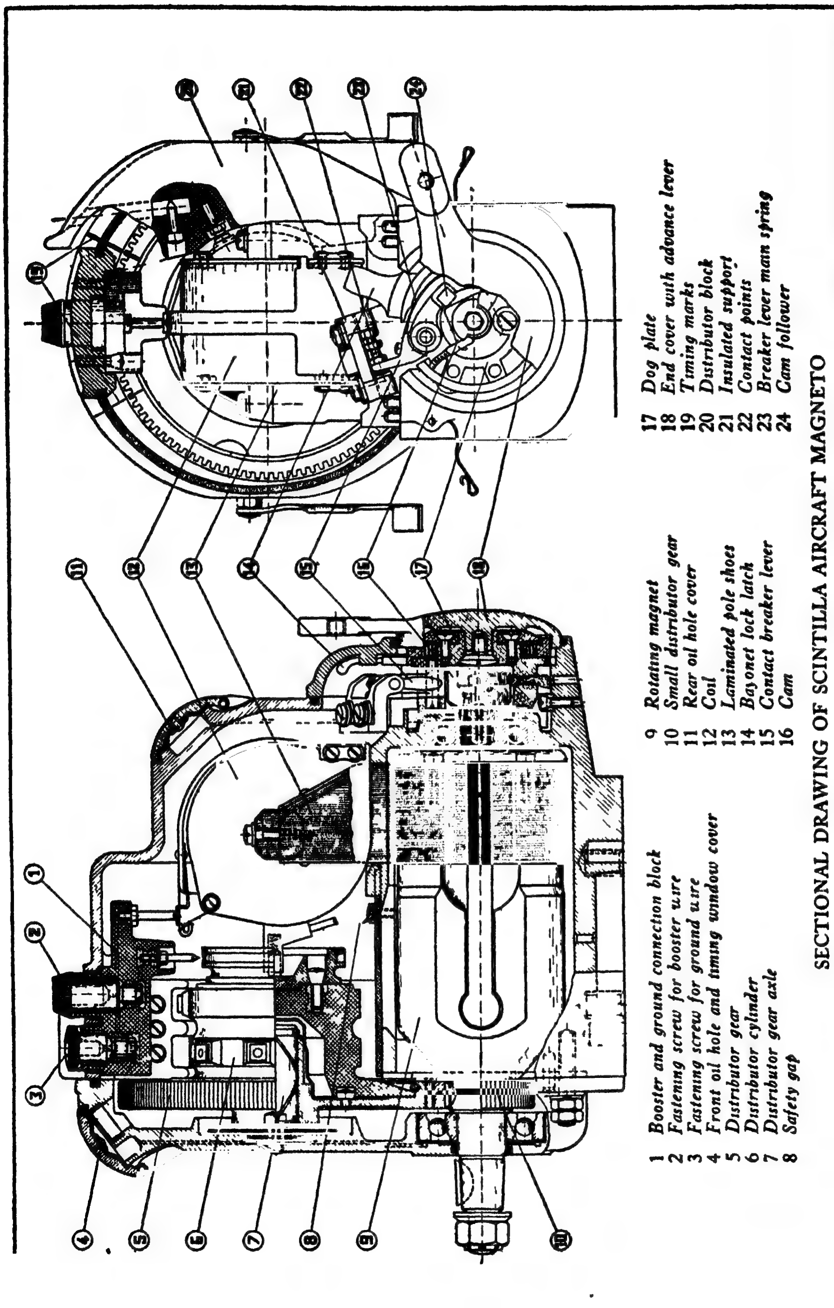


Fig. 179.—Sectional Drawing of Scintilla Aircraft Magneto Showing all Important Parts.

lever has a short reinforcing spring under the breaker lever end and a long reinforcing spring under the breaker cage end. It is fastened by a screw at each end. The end cover with advance lever is held solidly against the back of the contact breaker assembly by a screw which seats in the breaker cage and screws into the centrally located boss in the end cover. There are two dogs, 180 degrees apart which fix the position of the advance lever by engaging in holes in the dog plate. The advance lever may be fixed and held in any one of eight positions.

The Front End Plate.—The end plate is fastened to the main housing by two studs in the bottom holes and two screws in the upper holes. Its position is fixed by two dowel pins in the front end of magneto housing. The front end plate holds the outer ball race for the drive end bearing. It also carries the large distributor gear and distributor cylinder upon the distributor gear axle. The distributor gear axle is fastened to the end plate by two screws. The distributor gear is locked on its axle by a steel spring ring which seats in an angular groove in the end of the axle. The steel spacing washer between the back end of the distributor gear bearing and the spring ring on the axle provides a means of adjustment for the end play of the bearing. The distributor cylinder is locked to the distributor gear by a spring ring. The correct position of the distributor cylinder for a given rotation is fixed by a dog screw, which screws into the distributor gear. The spacing of the distributor cylinder from the gear by a large paper washer assures a tight fit for the spring ring against the distributor cylinder when the ring is pressed into its groove in the distributor gear. The front end plate also carries the distributor block spring clamps.

The Coil.—The high tension spark coil is mounted directly on the extensions of the laminated pole shoes, thus insuring the coil a maximum freedom from oil and grease as this mounting puts it well up under the main cover. The pole shoe extensions are ground to insure a good contact with the core of the coil. The coil is held in place by a screw in each end of the core.

The condenser is built in as an integral part of the coil. This assures protection for the condenser and a practically uniform capacity irrespective of temperature and moisture. The high tension carbon brush holder and safety gap electrode are mounted on the front of the coil. The ground connection and the spring contact for the insulated support of the stationary contact point are incorporated in the primary bridge which extends over the top of the coil and is secured by six small screws.

The Magneto Housing.—An aluminum housing covers the rotating magnet. It carries the outer race for the breaker end bearing, the ground plate for safety gap, the breaker cover spring clamps and the dowel pins for locating the main cover and breaker cover. The pole shoes are laminated and cast as an integral part of the magneto housing. The breaker stop and its fastening screws are located in the lower part of the breaker end of the magneto housing.

The Main Cover.—The main cover is located by four dowel pins and is fastened to the magneto housing by two screws. It affords protection to the coil from moisture, oil and dirt under abnormally severe operating conditions. The booster and ground connection block is mounted in the

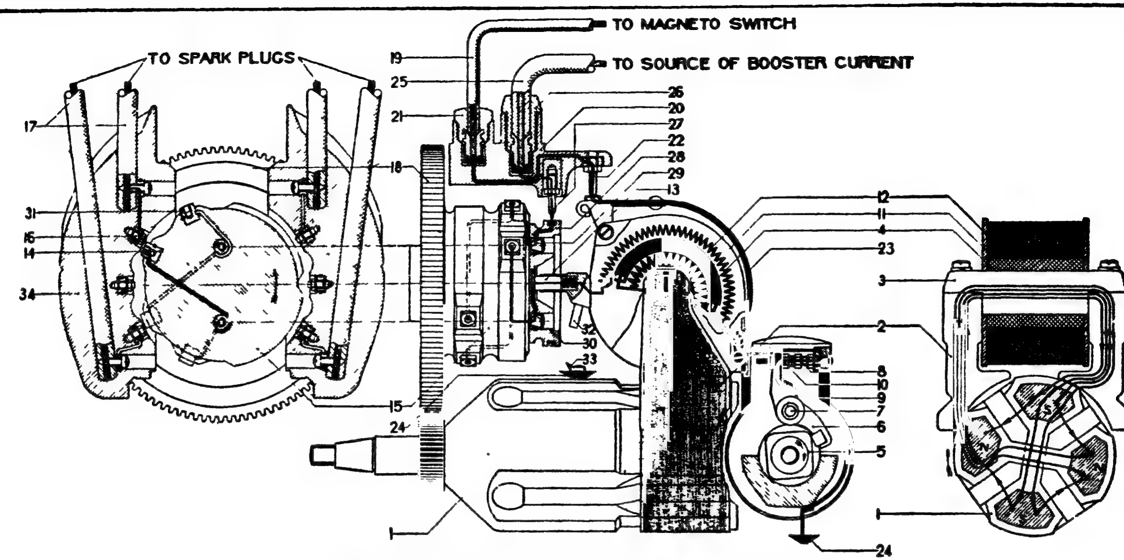


DIAGRAM OF ELECTRIC AND MAGNETIC CIRCUITS

1	*Rotating Magnet*	14	*Segment in Distributor Cylinder carrying Secondary Current*	25	*Booster Cable*
2	*Pole Shoes*	15	*Distributor Cylinder*	26	*Fastening screw for Booster Wire*
3	*Core of Coil*	16	*Electrode in Distributor Block*	27	*Electrode for Booster Current*
4	*Primary Winding*	17	*Ignition Cable*	28	*Collector ring for Booster Current*
5	*Breaker Cam*	18	*Large Distributor Gear*	29	*Fastening Screw for Collector Ring (in Booster Circuit)*
6	*Breaker Lever*	19	*Ground Wire*	30	*Fastening Screw for Collector Ring (in Secondary Circuit)*
7	*Breaker Lever Axle*	20	*Booster and Ground Connection Block*	31.	*Segment in distributor Cylinder carrying Booster Current*
8	*Long Contact Point (Insulated)*	21	*Fastening Screw for Ground Wire*	32	*Safety Gap Electrode*
9	*Main Spring for Breaker Lever*	22	*Stud for ground contact*	33	*Ground Plate for Safety Gap*
10	*Short Contact Point*	23	*Primary Bridge*	34	*Distributor block*
11	*Condenser*	24	*Ground through the Magneto and Engine*		
12	*Secondary Winding*				
13	*High Tension Carbon Brush*				

Fig. 180.—Diagram of Electrical and Magnetic Circuits of Scintilla Aircraft Magneto.

extension of the main cover between the distributor blocks. It is secured by two screws. The booster and ground connection block carries the ground terminal and the stud for ground contact, also the booster terminal and the electrode for the booster current. The stud for ground contact bears on a spring plate secured to the primary bridge. The electrode for the booster current is held directly over the collector ring for the booster current. There is a small air gap between the electrode and the collector ring. At the top of the main cover is provided numbers for locating the distributor blocks, an arrow showing the direction of rotation of the magneto and the two letters "H" and "P" to mark the Booster and Ground terminals respectively.

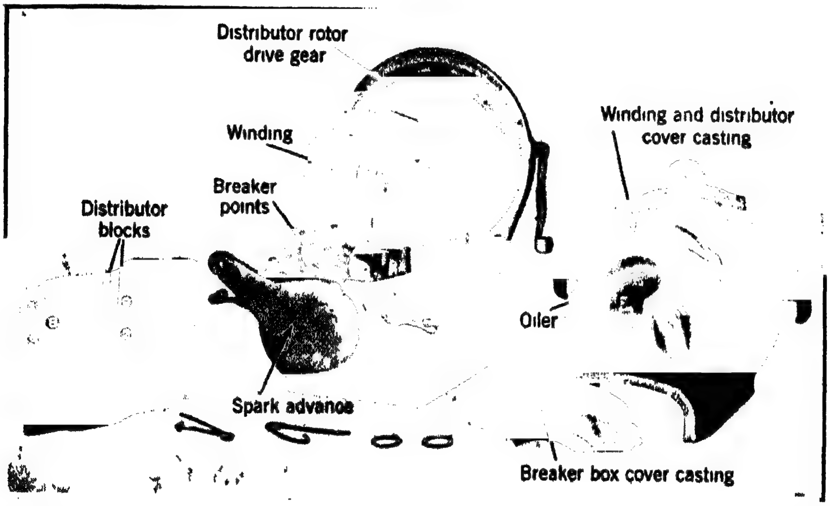


Fig. 181.—Scintilla Aircraft Magneto Partially Dismantled Exposing Interruptor Points and Coil.

The Distributor Blocks.—These are mounted so that they are held between the main cover and front end plate. Their lower ends rest upon the magneto housing while the upper ends fit against the top extension of the main cover. They are held in place by spring clamps and are designed as the Right and Left distributor block as viewed from the drive end. The breaker cover is located on the magneto housing by two dowel pins and held in place by a spring clamp at each end. It is directly over the contact points and its removal permits of a ready inspection of the points or removal of the contact breaker assembly.

Electrical Operation of Magneto.—This section is presented with the hope that the nontechnical reader may be able to form a better conception of the mechanical and electrical principles involved in the operation of the Scintilla Aircraft Magnetos and if studied in connection with the diagram at Fig. 180 the explanatory matter can be more easily followed. The rotating magnet (1) has four poles. The poles are joined together inside the laminated ends into pairs. The two "N" poles making up one pair and the other two "S" poles making up the other pair. The rotating magnet

(1) revolves between the laminated pole shoes (2) producing an alternating field in the core of the coil (3). When the current reaches its maximum value, the breaker cam (5) causes the breaker lever (6) to turn on its axle (7); thus opening the platinum contact points (8) and (10). The cam (5) is mounted on the rear end shaft of the rotating magnet (1) its position being fixed in relation to the magnetic field. The short contact screw (10) is connected to the ground (24) through the breaker lever (6) and the main spring for breaker lever (9) while the long contact screw (8) screws into the insulated support and maintains permanent contact

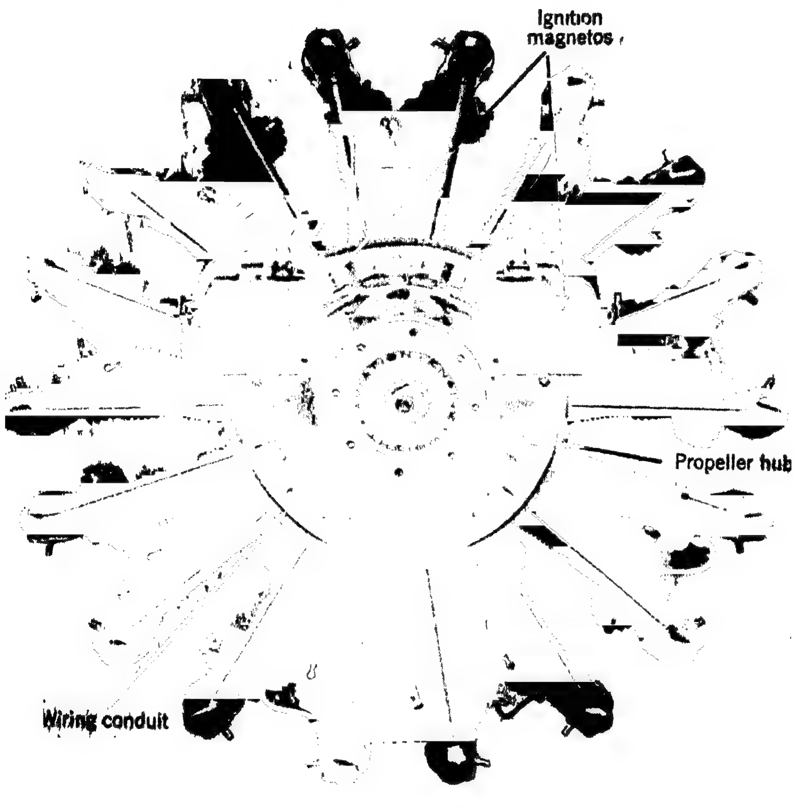


Fig. 182.—Wright "Whirlwind" Engine Viewed from Propeller End Showing Installation of Scintilla Aircraft Magnetos.

with the primary winding (4) by means of a laminated copper brush fastened on the primary bridge (23). Therefore, when the contact points (8) and (10) are open, the primary circuit current is suddenly interrupted. The condenser (11) is connected in parallel with the contact points (8) and (10). It prevents abnormal arcing at the points when the primary current is interrupted, thus reducing their wear to a minimum and insuring regular sparking.

High Tension Current.—The interruption of the primary current induces a high tension current in the secondary winding (12) composed of a large number of turns of fine wire. One end of the secondary winding (12) is connected to the ground (24) through the primary winding (4) and the core of the coil (3), while the other end terminates at the high tension carbon brush holder which is built in as an integral part of the coil. The high tension carbon brush (13) transmits the current to the sparkplugs through the medium of the distributor cylinder (15) and distributor blocks (34) and the ignition cables (17). The high tension brush (13) bears on the central contact of the collector ring for booster current (28) which is secured to the distributor cylinder (15) by two fastening screws (29) and (30). The screw (30) is located in the secondary current circuit and connects the central contact in the collector ring for booster current (28) with the conductor moulded into distributor cylinder (15) and leading to segment (14). The distributor cylinder (15) is fixed on the large distributor gear (18) in a definite position relative to the opening of the contact points (8) and (10) and for a given rotation which, in the diagram is anti-clockwise. Thus the segments (14) successively register with the electrodes (16) in the distributor blocks (34) thereby transmitting the secondary current to the ignition cable (17) and thence to the sparkplugs.

Safety Gap.—The safety gap is the space between the insulated electrode (32) which screws into the high tension carbon brush holder and the electrode (33) on the safety gap ground plate. Its function is to protect the coil against excessively high voltage by providing a means of escape for the charge, which will jump the gap between the electrodes (32) and (33) in the event of the secondary circuit being accidentally broken between the plugs and the coil.

The advancing and retarding of the ignition is obtained by moving the breaker assembly about the cam (5). Moving the breaker assembly against the direction of rotation of cam (5) gives advance, while moving breaker assembly with direction of rotation gives retard.

Booster Connection for Starting.—The booster cable (25) is held by fastening screw (26). The booster current is carried to the electrode for booster current (27) through the medium of a conductor moulded into the dielectric material of the booster and ground connection block (20) thence through a small air gap to the collector ring for booster current (28). The fastening screw for collector ring (29) located in the booster current circuit, transmits the booster current to the segment (31) in the distributor cylinder (15). The booster current then jumps the air gap to the nearest electrode in the distributor block (34) and thence through the ignition cable (17) to the sparkplug. The booster current segment is located in such a manner that it trails the secondary current segment (14).

Stopping the Engine.—To stop the engine, the ignition is cut out by neutralizing the functioning of the contact points (8) and (10). This is accomplished as follows: The end of the primary winding (4) terminates through the spring contact on top of the primary bridge (23) and thence through the stud for ground contact (22) to the primary terminal marked "P" and carrying the fastening screw for ground wire (21). The ground

Dimensions in Inches

	AG-8 D	AG 9 D AG 12-D
A	1.968	1.968
B	8 063	8.063
B1	7 206	7.205
B2	6.476	6 476
C	1.693	1.693
D	2.519	2.519
E	.500	500
F	591	591
G	2.087	2.087
H	1.969	1.969
I	.787	.787
J	2.362	2.362
K	.235	.235
L	8.031	8.031
M	625	.625
N	8.581	8 581
O	.925	925
O1	2.756	2.756
P	4.390	4.390
R	.313	.313
S	3.150	3.150
S1	3.228	3.228
T	5.313	5.813
U	1.969	1.969
V	.125	.125
W	.062	.062
X	⅜-16	⅜-16
Y	.591	591
Y1	.472	.472
Z	⅜-16	⅜-16
Z1	.296	.296

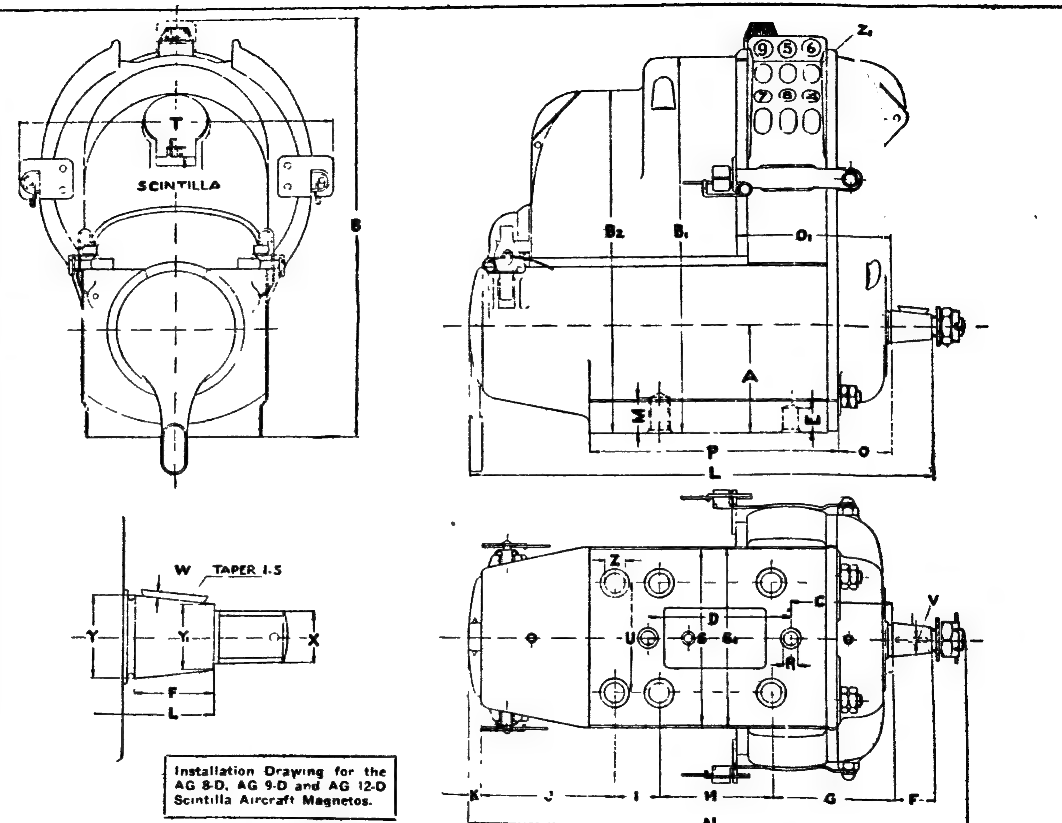


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Fig. 183.—Installation Drawing for the AG-8D, AG-9D and AG-12D Scintilla Aircraft Magnetos.

wire (19) goes to a switch located conveniently for the pilot. When the switch is closed, the effect of the contact points (8) and (10) is neutralized by permitting the primary current to flow around the points and through the switch to the ground thus grounding the primary current and causing the ignition to the engine to cease.

Taking Down Scintilla Magneto.—The following procedure is advised by the makers in their instruction manual for dismantling the magneto in a systematic manner:

(1) Remove the safety pin on breaker cover and distributor block spring clamps.

(2) Release spring clamps and remove breaker cover by lifting straight up.

(3) Remove breaker assembly. This is readily accomplished by moving the breaker to midway between advance and retard positions. Now bring the bayonet lock hand latch to a vertical position; this unlocks the bayonet lock and the breaker assembly is easily removed by pulling outward.

(4) Release spring clamps and remove distributor blocks.

(5) Unscrew fastening screws for magneto cover and remove it by lifting straight up until cover clears coil. Should the cover be tight on the dowel pins, it can be loosened by alternately tapping it with a fiber drift, on the rear edge just over the breaker compartment, and lifting the front by hand.

(6) Unscrew fastening screw in each end of core of coil and remove coil by pulling it back until the high-tension brush clears the distributor cylinder; then lift coil out. Care should be exercised in removing the coil. Do not pull straight up on it until the high tension brush clears the face of the distributor cylinder. Some coils fit fairly tight between the pole shoe extensions and when they release themselves under a vertical pull, will do so quite suddenly and the high-tension brush will be broken and possibly its holder torn loose from the coil; hence the injunction—move coil back until the high-tension brush clears the face of the distributor cylinder, then it may be lifted out.

(7) Remove the two fastening screws and the two nuts and lock washers from each of the bottom studs, then pull the front end plate off. If the end plate should fit tight, it may be loosened by alternately tapping it gently on each side of the inside surface with a rawhide mallet.

(8) Remove the rotating magnet. This is easily accomplished by turning the rotating magnet until an opening between any two poles of the magnet appear in the center between the top edges of the pole shoes. This allows the flat surfaces of the magnet to be in such a position that the rotating magnet can be readily withdrawn from the magneto housing.

(9) Take distributor cylinder off the large distributor gear by releasing the spring ring that holds it. There is a recess in the outer edge of the distributor cylinder flange for the ends of the spring ring. The ring may be released by prying with a small screwdriver, between

the flat part of the ring and the inside of the distributor gear, thus forcing the end in far enough to clear the groove cut for it. When ring is released, the distributor cylinder may be lifted off.

(10) Remove large distributor gear by first releasing the small steel spring ring, then removing the spacing washers and lifting gear off the axle shaft.

(11) Remove end cover with advance lever from breaker assembly, by unscrewing fastening screw in back of breaker cage.

Cleaning Scintilla Parts.—All parts may be washed in gasoline and dried with compressed air except the coil. Wipe all pieces of dielectric material with oil-saturated cloth after cleaning. There are, however, several precautions to be observed in drying off the parts. The number discs in the distributor blocks and those on the top of the main cover must not be exposed to full air pressure. Hold them at a safe distance or else allow them to dry in the open air. Great care should be taken that the felt strips in the magneto housing, the main cover, the breaker cover and the front end plate are not loosened or torn out by the air pressure. Oil all felt strips after cleaning.

It is imperative that the cage and bearing assembly of each bearing be held so that they cannot spin when using air for cleaning. This will prevent throwing out the balls and making it necessary to obtain a new cage and ball assembly to complete the bearing for re-assembly. Keep rotating magnet clean inside and out. Do not lay it near small screws, nuts or metal chips, etc. Its construction is such that any foreign material adhering to it will result in serious injury to the magneto. After cleaning rotating magnet, grease thoroughly to prevent rust.

Inspection of Scintilla Magneto.—It is anticipated that the inspection and repair of the Scintilla Aircraft Magneto will offer very little, if any, difficulty to the average ignition mechanic, who has had experience with other types of magnetos. Therefore, this section on inspection and repair will embody only such instructions as are deemed necessary to cover certain features which are characteristic of the Scintilla Aircraft Magneto.

Magneto Housing.—The breaker stop and the safety gap ground plate must be tight and their fastening screws locked. Have 3/8-16 cap screw hold down holes clean and threads straight. Lap base just enough to see that its surface is smooth. It is imperative that the oil lead to back bearing be open and clean. Flush with a good grade of light oil after cleaning.

Front End Plate.—Lift oil wick and spring out of distributor gear axle. Clean oil leads to axle and front end bearing and flush with a good grade of light oil. Replace oil wick in distributor gear axle. See that fastening screws for distributor gear axle are tight and locked to the outside surface of the front end plate. Examine large distributor gear to see that there are no burrs on teeth of gear. Replace gear, taking care to hold wick down until covered by gear bearing. Replace spacing washers and spring ring. Try end play of gear on shaft. If not less than .005 inch or more than .008 inch, it is satisfactory. Test dog screw. It must be tight and locked to the large distributor gear.

Replace paper spacing washer and distributor cylinder. Replace spring ring. Total thickness of spacing washer should be such that distributor

cylinder will be held tightly against distributor gear. Force spring ring into its groove throughout its length.

Breaker Assembly.—Replace end cover with advance lever on breaker assembly. Place assembly in position in magneto housing and note that it functions as follows:—The bayonet lock latch, when released, should snap into position and the breaker will spring over to full advance. Remove breaker and lay aside for adjustment during final assembly. .

Main Cover.—Clean oil lead to back bearing thoroughly. Examine booster and ground connection block in top of main cover, especially around the terminal marked "H," as any small cracks in the material would ground the booster current.

Coil.—Note that secondary brush holder is solid with the coil. It is of vital importance that the spring contacts on the primary bridge be in good condition. The rear spring bears against the face of the insulated support on top of the breaker cage, while the front spring located above the coil, makes contact with the ground contact stud.

Distributor Blocks.—Examine electrodes. Be sure that they are screwed tight into the distributor block. Loose number discs must be glued with a water and oil proof glue. After glue is dry, apply white shellac as an added precaution.

Rotating Magnet Assembly.—Check cam fastening screw. Note condition of ball bearings. It will be noticed that the cage and balls of the front bearing are a loose part on the AG 12-D rotating magnet, while on the AG 8-D and AG 9-D rotating magnets they stay on the inner race. This allows the balls to clear the large distributor gear during the assembly of the AG 12-D. All types of rotating magnets carry the cage and balls for the rear bearing on the inner race. Examine laminated pole end of magnet for any signs of rubbing due to foreign material lodging between laminated ends of magnet and pole shoes. The clearance between laminated poles and pole shoes is .002 of an inch. This explains the necessity of keeping them clean and free from any foreign material.

Assembly of Scintilla Magneto.—It is presumed that as near as possible the mechanic will use the original parts of the magneto for the re-assembly. While Scintilla parts are readily interchangeable in each type of magneto and for a given rotation will function in another magneto of the same rotation, much time and effort will be saved by using the same rotating magnet, magneto housing and front end plate in the re-assembly. The end play and bearing fit of the rotor in the magneto in most instances will be found correct. The rotating magnet was taken as the last sub-assembly for inspection so that while it was cleaned up it could be installed in the magneto housing immediately.

(1) Have magneto housing clean and ready to receive rotating magnet.

(2) Take up the rotating magnet and fill the rear ball cage with good light grease. Grease magnet all over, leaving a film of grease to prevent rust.

(3) Recharge rotating magnet, clean off metal particles that may be adhering to poles and place in housing at once. The magnet is easily replaced by turning it until a flat surface is at the top, then push in place. Now turn rotating magnet right or left 45 degrees

or until the space between the top of the pole shoes is filled by one of the poles of the magnet. This is the neutral position for the rotating magnet and it should always be left in this position unless there is a keeper across the pole shoe extensions.

(4) Fill cage and ball assembly for front bearing with the light grease mentioned above, put it on over the shaft and place it on the inner race. **NOTE: If the magneto is an AG 12D the cage and ball assembly must be placed in outer race in front end plate and assembled with it.**

(5) Observe the arrow on top of the main cover to find direction of rotation for which internal timing was originally set. If arrow points anti-clockwise, as viewed from the drive end of the magnet, match all timing marks "G." If arrow points clockwise, match all timing marks "D." Suppose the magneto to be assembled is an anti-clockwise or left hand rotation:

(6) Turn rotating magnet until the marked tooth on back of small distributor gear is up in view so that it may be matched with the marked tooth on the large distributor gear.

(7) Take up front end plate and put it on over drive end shaft until edges of gears are about to touch, holding the plate in one hand and guide the mark on the large distributor gear by turning the distributor cylinder with the other hand. When the marked tooth on the small gear and the marked tooth on the large distributor gear are matched, push the front end plate up against the magneto housing and secure by means of the two screws and studs provided.

(8) Test rotating magnet for end play. There should be none. The bearings should be just tight enough that when the magnet is turned about 30 degrees from the neutral position it will return to the neutral position by its own magnetic pull.

(9) Replace breaker assembly for final setting. Set contact points so that their maximum opening will be .012 of an inch. The small gauge on the Scintilla contact point wrench may be used for this purpose.

When contact points are set at .012 for maximum opening the clearance between back of breaker arm and face of fiber stop should be from .002 to .010 of an inch.

Check clearance on each cam lobe. The cam must run true within 0.0005 of an inch.

(10) The internal timing of the magneto must now be checked. Turn rotating magnet until Fig. 1 on large distributor gear is in line with the mark in the timing window; the supplemental timing marks, located on inside edges of large distributor gear and the front end plate, should also be in line at this time.

By slowly rocking rotating magnet with the breaker in full advance position, the points should be just on the instant of opening as Number 1 and its mark and the supplemental timing marks come in line with each other. Hold rotating magnet with timing marks

in line and with one hand place the right distributor block in position. When magneto is correctly timed, the Number 1 electrode will coincide with a segment on the distributor cylinder.

(11) Remove breaker assembly; this permits an easier installation of the coil.

(12) Place coil between pole shoe extremities. This is best accomplished by sliding coil in from the back and moving it forward into position. The coil fits tight and often causes the pole shoe extensions to shear off a very thin piece of the fiber side plate. Take every precaution that none of this fiber gets in between the ends of the core of the coil and the ends of the pole shoe extensions. Secure coil with a fastening screw in each end of the core.

(13) Replace breaker assembly. Spin magneto; if properly assembled and timed a good snappy blue spark will jump across the safety gap. The safety gap should be not less than 3/8 or more than 1/2 of an inch.

(14) Put main cover in place. Take great care that it fits housing. Have bottom edges of main cover smooth. It is important that cover fits housing accurately since the top extension of cover acts as a stop for the distributor blocks while the housing supports them at their lower end. Any serious mis-alignment would result in injury to electrodes in the distributor blocks and segments in the distributor cylinder. Fasten main cover to housing with two long screws provided.

(15) Replace breaker cover. Fasten spring clamps and safety.

(16) Replace distributor blocks. Match them up with the number discs on the sides of the top of the main cover. Fasten distributor block spring clamps in place and safety.

This completes the final assembly.

Testing Magneto After Assembly.—Where the equipment is available, the magneto should be tested immediately after final assembly is completed. Its operation during the bench test gives the experienced mechanic an accurate knowledge of the mechanical and electrical condition of the magneto. Mount magneto on test bench and remove breaker assembly. Run magneto up to about 1,000 r.p.m. and listen carefully to its running. The period of any unusual or irregular noise as compared with drive shaft speed, will give a good indication as to its origin.

While listening to the magneto, note the hum of the gears. When running properly they will have a consistent hum. The pitch of this hum will vary slightly for different magnetos, but will be consistent for a given magneto when gears are running properly. When there is foreign material imbedded in the face of the large gear, it will cause an audible click or knock each time it is turned against the small distributor gear. When foreign matter is imbedded in the face of the small distributor gear, the period of the knock will be that of the drive shaft rotation since the small gear is keyed to the drive shaft. Should the gears run irregularly and tend to chatter, there is excessive play either in the distributor gear axle bearing or between the teeth of the gears. If there is a knock in the magneto housing, it is usually caused by the laminated poles hitting the pole shoes.

Most trouble in the magneto housing is caused by small metal parts, such as filings, chips, broken lock washers, etc., that are picked up while the rotating magnet is lying on the bench. They are thrown out by centrifugal force and cause serious damage to the rotating magnet and pole shoes. Unusual tightness or rubbing in housing may be noticed by the magneto running exceptionally warm during test. If the condition is bad, it will be noticed by an irregular and unusually hard turning, when the drive shaft is turned by hand prior to the test run.

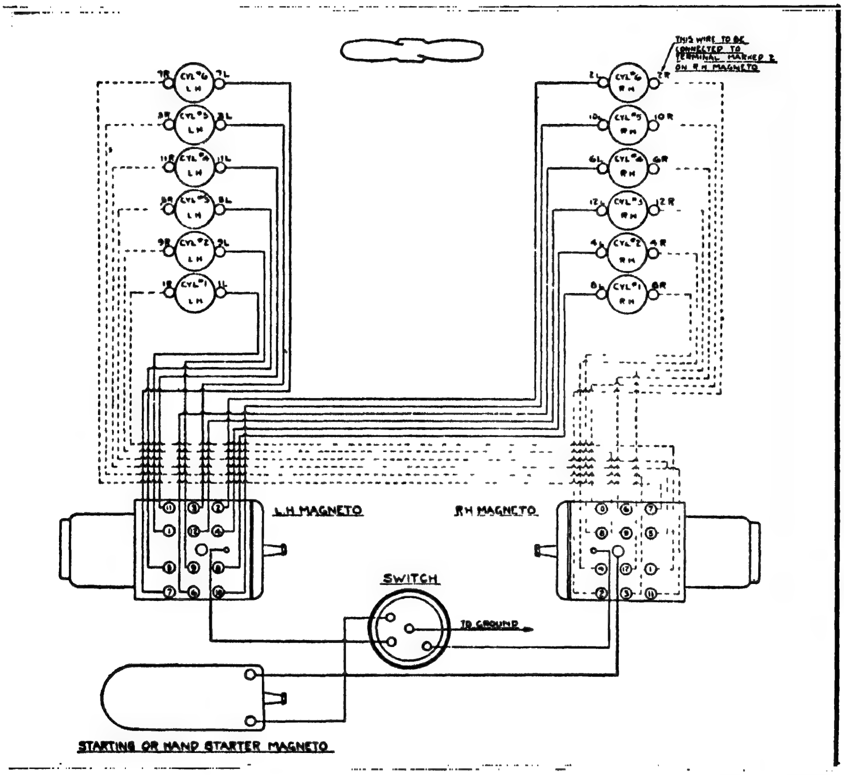


Fig. 184.—Wiring Diagram of Curtiss D12 Scintilla Magneto Ignition System.

Electrical Tests of Scintilla Magneto.—Replace breaker assembly and wire up distributor blocks to spark gap, which should be of the three electrodes type in order to obtain good and consistent results for testing. The third electrode is a static point located so that it meets the live point at an angle of 90 degrees and is set with an air gap of about .002 of an inch between static point and live point. The live point is the one connected to the distributor block by the high tension cable. The gap between the live point and grounded point is to be seven millimeters ($\frac{9}{32}$ of an inch). Keep all points sharp and clean to obtain best results. Run magneto for about ten minutes at a speed of 900-1,000 r.p.m. Observe points at start of test. Should they arc excessively, remove breaker and clean them thoroughly. Have points dry and free from grease before replacing breaker.

After ten minutes at above speeds, run magneto up to 2,800 r.p.m. and maintain this speed for at least five minutes. Observe the contact points. There should be very little sparking at points when they are clean and seat properly on their contact area. Test primary grounding circuit at this stage of the test with a piece of wire.

Put one end in the hole in fastening screw for ground wire and touch the other end to magneto body or frame of test stand. The spark across the gap should cease the instant the free end of grounding wire is touched to magneto or frame of test stand. The next running speed should be not

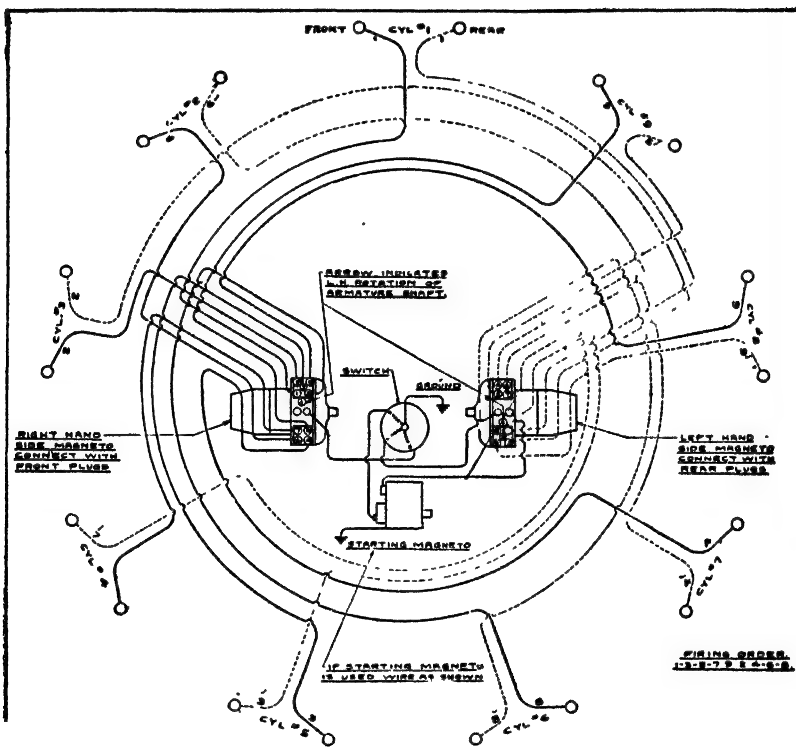


Fig. 185.—Wiring Diagram Showing Application of Scintilla Magnetos to Wright "Whirlwind" Model J5 Engines.

less than 3,500 r.p.m. while 4,000 r.p.m. is desirable. Maintain this speed for at least ten minutes. Observe spark closely. A consistent miss can readily be detected. A slight scattering miss will be detected by a momentary break in the spark flame between the electrodes. A magneto, the internal timing and assembly of which is correct, will regularly fire across a seven millimeter air gap at 4,000 r.p.m. Observe contact points. They should run practically free from arcing by the time the test has progressed thus far.

The next test, after the high speed test, is to obtain the coming-in speeds. The coming-in speed is the lowest speed at which the magneto

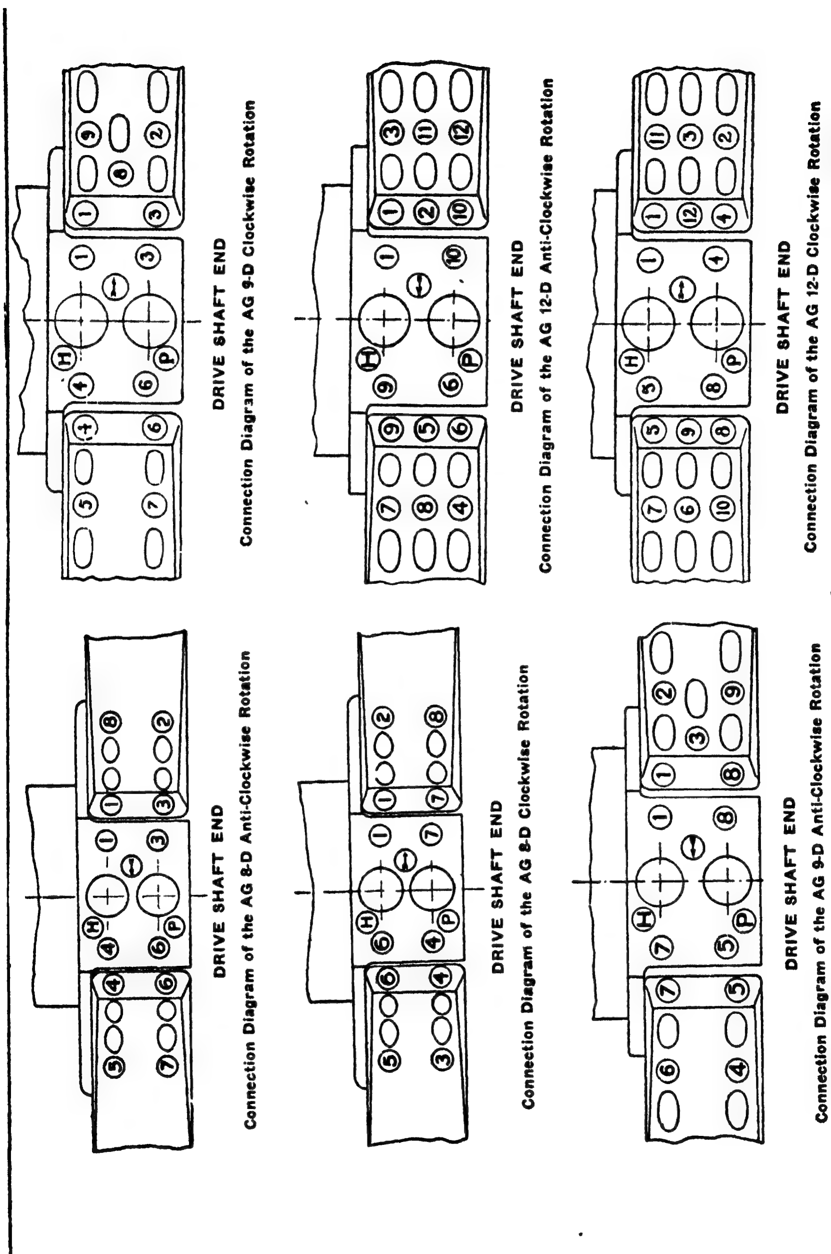


Fig. 186.—Distributor Block Arrangement of Various Scintilla Magneto Models.

will come in and fire consistently across the seven millimeter air gap. The maximum coming-in speed allowed for full advance is 120 r.p.m. while that for full retard is 240 r.p.m. Coming-in speeds higher than those just given would indicate a magneto of low electrical efficiency. This may be due to improper internal timing or to the rotor having lost some of its magnetism through improper handling after being recharged or by spinning the rotating magnet without a keeper across the pole shoe extensions.

The booster starting circuit should be tested after the finish of the running tests. Use a hand magneto or vibrating high-tension coil as the source of high-tension current. Connect high-tension booster current cable to terminal marked "H." Rotate magneto by hand. Start with Number 1 in the timing window and observe that a spark jumps all gaps in the correct serial order as given by numbers on distributor blocks. Bear in mind that because the booster segments trails the service segment of the distributor cylinder, the booster spark will occur late, for example; when Number 1 shows in the timing window, the booster current will jump Number 12 gap. The same principle applies to the eight- and nine-cylinder magnetos. A deviation from this serial order or no spark at all across gap indicates that the booster circuit is not correct. Examine parts for fine cracks or burnt spots indicating a short circuit of the booster current.

Charging Magnets.—The Scintilla Plunger Type Magnet Charger is designed to charge either a two or four pole magnet. There are two control switches used. One is a hand switch, and the other is operated by foot pressure. The source of charging current should be a twelve-volt storage battery. The hand switch has two points marked with a Number 2 and Number 4 respectively. The magnet is inserted in a sleeve and the sleeve is moved in and out of a strong electro-magnetic field by hand. The sleeve is locked to its handle by a bayonet lock. The handle is fixed in position so that when charger is in operation, it can be moved only in a vertical plane.

To Operate.—Lift up handle with sleeve attached. Unlock sleeve and insert magnet, using care that the magnet pole surfaces are covered by the iron inserts in the sleeve. The relations between the N and S poles of the magnet and the polarity of the charger may be disregarded. Lock sleeve to handle and replace in charger in original position. Assume that the magnet to be charged is a four pole type. Move hand switch to point marked with the Number 4; close charging current circuit by pressing the switch on base of charger with the foot. Move handle locked to sleeve containing magnet up and down at least twenty times while the charging circuit switch is held closed with the foot. Do not move the sleeve in and out of the electro-magnetic field too fast. Use a rate of about one second for each count. When charging is completed pull handle with sleeve attached out of charger and remove magnet. It is imperative that the charging circuit switch at base of charger be held closed while the sleeve containing rotating magnet is being removed from charger. Place magnet in magneto housing in the neutral position as soon as possible.

Installing Scintilla Magneto.—Make sure that the magneto shaft half of the drive coupling is seated and keyed on taper of drive shaft and then

locked by nut and washer and safetied with a cotter pin. Spin magneto and note that drive shaft half of coupling runs true. Inspect magneto base. See that the ⅜-16 threads in the hold-down holes have not been stripped or that the start of the thread has not been closed; thus having a tendency to cause cap screw to cross threads. When necessary, clear up threads with a ⅜-16 tap and then clean holes thoroughly. Note dowel pin holes. They should fit pins very snugly but not so tight as to cause difficulty in getting magneto down on surface of its support. See that the base of the magneto is smooth and makes good contact with the bracket surface. Having made use of the above instructions and any other information that the mechanic has found to be useful from previous installations of this nature, the magneto is ready to be secured. It is imperative that the length of the ⅜-16 cap screws be correct. When in doubt as to correct length, it can best be obtained by direct measurement. Making allowance for the thickness of the washer used on the screw, the length must be such that when the screw is tight, it will not have less than 7/16 or more than ½-inch of its threaded length in the ⅜-16 hole in the base of the magneto. When the correct length is established draw the screws up tight and safety.

Timing Scintilla Magneto.—Turn engine slowly as piston in cylinder Number 1 comes up on compression. Stop when the full advance position for the ignition, as given by the engine manufacturer, is reached. This point is given in degrees before top center (B.T.C.) and is marked on the timing disc to be used with the engine. At this point the magneto is to be coupled to the engine, the following instructions having been very carefully noted and followed. The Number 1 on the large distributor gear can be seen through the timing window located under the front oil hole cover. When this Number 1 is in line with the white mark at the top of the timing window it indicates that the contact points are at the instant of opening with the breaker fully advanced and the Number 1 electrode on the distributor block is registering with the proper segment on the distributor cylinder. Some installations do not permit of this window being used. In such cases the supplemental timing marks must be used. They are located on the inside surface of the large distributor gear and the front end plate and are so arranged that when they coincide with one another the contact points are at the instant of opening.

Fasten the sparkplug cables to the distributor blocks. The numbers on the distributor blocks show the serial firing order of the magneto. The numbers on the top of the main cover are for the purpose of locating the right and left distributor blocks to their respective sides. Observe that each magneto is wired to its proper set of wires as given by the engine manufacturer and that the wires lead from the magneto so that Number 1 on the distributor block goes to Number 1 or first cylinder in the firing order of the engine, while Number 2 on the distributor block goes to the second cylinder in the firing order of the engine; Number 3 on the distributor block goes to the third cylinder in the firing order of the engine, etc., until all cylinders have been wired in their proper firing order. Clamp distributor blocks in place. The letter "H" marks the terminal to which the booster wire is to be connected. The letter "P" marks the terminal

to which the ground or short circuiting wire from the magneto switch is to be connected. The arrow indicates direction of rotation of the magneto when viewed from the drive end. Tighten and lock the drive coupling. The advance lever linkage is to be connected to the advance lever on the magneto, special attention being given that full advance and retard are obtained when the spark lever in the cockpit is moved to its full advance and retard positions.

Changing Direction of Rotation.—Disassemble magneto. Pull the cam and observe the small "D" and "G" stenciled on the face of the breaker end shaft. Assuming that this is to be a change of rotation from anti-clockwise to clockwise, remove Woodruff key from keyway marked "G" and replace in the one marked "D." Replace cam and pull cam fastening screw up tight. Remove distributor cylinder and then the large distributor gear. Remove small dog screw from "G" (within the circle) and replace in threaded hole marked "D." Lock end of dog screw in face of distributor gear. Note that "G" (without a circle around it) is to be used only when the magneto is for anti-clockwise rotation *and has no booster connection.* Remove the timing window and the supplemental timing marks on the inside surface of the front end plate. Replace large distributor gear and distributor cylinder. The distributor cylinder is now in the correct position for right hand rotation. Observe that the collector ring for booster current has a "D" and "G" on its face, located 180 degrees from each other. There is a mark on the inside face of the distributor cylinder with which either the "D" or "G" must line up, depending upon the direction of rotation. Remove collector ring for booster current and turn it so that "D" is in line with the mark on the inside face of distributor cylinder. Replace and fasten collector ring for booster current to distributor cylinder with fastening screws provided for it.

Remove end cover with advance lever from breaker assembly. Unscrew dog plate fastening screws and remove dog plate. Now carefully lift the bayonet lock latch and spring off breaker cage. Remove the bayonet lock spring from the bayonet lock latch, turn it over and put it back in place. This is done in order that the lock spring may hold the breaker assembly in full advance. The contact breaker lever axle and breaker lever are to be removed. This permits of an easy exchange of locations for the fiber stop and the long contact screw. Remove long contact screw and fiber stop and replace each one in the opposite hole. Reverse contact breaker lever and replace so that the contact points match. Screw breaker lever axle in tight and lock threaded end to breaker cage. Now replace bayonet lock latch and spring. Note that each end of the spring is secured. One end fastens in bayonet lock latch while the other or inside end fastens in breaker cage. Replace dog plate and secure with dog plate fastening screws. Lock screw heads to dog plate. Fasten end cover with advance lever in place and the breaker assembly is ready for clockwise rotation.

Recharge rotating magnet; note that it is clean and free from clinging metal particles and replace in magneto housing. Turn magnet until marked tooth on distributor gear is up. Take front end plate and put in position for matching gears. Turn large distributor gear until tooth marked "D" lines up with marked tooth on small distributor gear. Mesh gears and slide

front end plate into place and fasten to magneto housing with nuts and screws provided. Replace contact breaker assembly. Turn rotating magnet until the number "I" on the distributor gear appears in the opening for the timing window in the front end plate. Now hold the rotating magnet so that the contact points are just at the instant of opening and then replace the timing window so that the white mark therein will coincide with the mark on the large distributor gear directly above the number "I." Remark the front end plate with new supplemental timing marks that will coincide with the original marks on the distributor gear. The new marks will be correct for clockwise rotation of magneto.

Replace main cover and change rotation indicating arrow from anti-clockwise to clockwise rotation. Change number discs to correspond with clockwise rotation as shown in distributor block diagrams and replace distributor blocks. The magneto is now ready for test as a clockwise (R.H.) rotating magneto. If it is required that the direction of rotation of the magneto be changed it is strongly recommended that the magneto be sent to the Scintilla factory for this purpose.

Adjusting End Play of Rotating Magnet.—As there is only .002 of an inch air gap between the rotating magnet and the pole shoes, and in consideration of the design of the ball bearings, it is imperative that there be a very careful adjustment of the axial or end play of rotating magnet between its bearings. Since the rotating magnet exerts a certain amount of attraction between its poles and the pole shoes, when turned from the neutral position, advantage is taken of this pull to get the correct adjustment of the ball bearings. The end play adjustment is really carried beyond the point where a mechanic could actually feel even the slightest hint of end play. The correct adjustment is observed by turning the rotating magnet away from its neutral position and noting how far the trailing edge of the rotor slot can be from the edge of the opening between the pole shoes and still return to its neutral position. Obviously, the closer the edge of the slot must be to the pole shoe edge in order to pull back to neutral, the tighter the bearings.

It has been found in practice that when the distance between the pole shoe edge and the rotor slot edge is from 3/8 to 3/4 of an inch, the adjustment is satisfactory, while 5/8 of an inch is the desirable adjustment. The adjustment of the end of the play is obtained by means of steel spacing washers which fit in between the inner ball races and the rotating magnet. Keep the total thickness of spacing washers just as nearly equal as possible on each end. These spacing washers come in thicknesses of .05, .1, .2, .3, .4 and .5 of a millimeter, which is approximately .002, .004, .008, .012, .016 and .020 of an inch respectively.

Installing Outer Bearing Races.—The outer bearing races in the Scintilla Aircraft Magnetos are insulated from the magneto by insulating strips of a specially prepared material. They are also backed by washers of this material which is very rigid and will stand considerable pressure. Thus the ball bearings are free from minute electric arcing which causes the balls to turn black and promote excessive wear. These ball races are readily removed by a specially designed puller and are easily pressed in by another tool. There is a recess cut in the metal where the outer races fit. This

recess allows the overlapping of the ends of the insulating strip. There is another recess cut in the metal back of the insulating washer; it is for oil. In installing an outer race, first put the washer in so that the cut in it will line up with the oil recess. Then spread a few drops of oil evenly over one side of the insulating strip. Take strip up and bend it in a circular form, with the oiled side inside, and overlap the ends enough to allow the strip to fit easily into the container for the outer race. When it is released, it will expand against the container walls and the ends will overlap slightly in the recess cut for them.

Place outer race squarely inside the insulated strip and press until race seats against insulating washer. The race should have a light press fit which may be obtained by the selection of insulation strips of the proper thickness. Great care must be exercised in keeping in line the race and the front end plate, or the race and the magneto housing as the case may be, so that the race will not cut the insulating strip, or, by not being seated squarely, it would result in serious injury to the ball races or the cage and ball assembly.

Contact Points.—The life of the contact points depends to a great extent on how clean they are kept. Therefore, take particular pains to keep them as clean as possible. File them only when it becomes absolutely necessary for the best operation of magneto. The gap between contact points when fully opened should be maintained at .012 of an inch. Use the gauge on the Scintilla wrench for this purpose. Keep points in alignment.

Adjusting Fiber Stop.—The fiber stop is mounted in the insulated support just back of the top of the breaker lever. Its functions is to limit the travel of the breaker lever at high magneto speeds. The fiber stop clearance is measured between the face of the stop and the back of the breaker lever. The clearance must be measured with the points fully opened. The fiber stop clearance must not be less than .002. This may be obtained in a practical way by using a .002 feeler, placing it behind the breaker lever and turning rotating magnet until the points are at their maximum opening. The tension on the feeler, as it is pulled from between the back of the breaker lever and the fiber stop, should be just enough to allow it to be withdrawn easily.

Care should be taken in filing fiber stop so that when proper clearance is obtained and the breaker lever is pushed back against the stop the whole surface of the stop bears against the back of the breaker lever. It is impractical to state a maximum fiber stop clearance which could be applied impartially to all Scintilla Aircraft Magnetos, in as much as the fiber stops and cam followers and contact points do not wear exactly alike.

Distributor Block Electrode Clearance.—There will be very little necessity for putting in new electrodes in the distributor blocks. However, should it become necessary, it is very important that they have the correct clearance. The minimum clearance should be .030 of an inch. The maximum clearance should not be over .050 of an inch. Templates for measuring these clearances may be obtained from the manufacturer.

Adjusting Mesh of Large Distributor Gear.—This adjustment will rarely be found necessary. In the event that the large distributor gear

requires replacement, adjust gear as follows:—To raise gear, loosen fastening screws on distributor gear axle flange and drift axle slightly to the right as viewed from the distributor gear side of the front end plate. To lower gear, drift axle slightly to the left. Tighten fastening screws for distributor gear axle and lock outer ends to outside surface of front end plate.

Timing Magneto by Means of Lights.—This practice can be followed very easily while magneto is on the repair bench and the coil is not installed, and will serve as a very accurate means of checking the internal timing of the magneto. Once the coil is installed, however, and the magneto is mounted on the engine, the timing marks on the magneto will be found sufficiently accurate to meet all practical requirements for timing magneto to the engine. With coil in place, the internal circuits and construction of the magneto are such that if an external source of current is applied to the contact points, there is the danger of the coil acting as an electro-magnet and weakening the magnetic qualities of the rotating magnet; thus impairing the efficiency of the magneto.

Oiling the Magneto.—Proper oiling is of vital importance for the satisfactory operation of the magneto. Use the best grade of medium bodied oil obtainable. Put from 30 to 40 drops of oil in the front oil holes or until it appears at the overflow hole for the distributor gear axle bearing. This hole is located about one inch below front oil hole cover. Put about three to five drops of oil in the back oil hole. Note that the felt wick in the bottom of breaker cage is thoroughly saturated with a heavy bodied oil. Oil magneto thereafter at regular intervals of about ten hours.

It is essential that the work of general overhaul and repair of the magneto be accomplished with the proper tools. The Scintilla Company, as a result of many years of experience, knows the necessity of removing the breaker cam, small distributor gear, ball races, etc., with tools specially designed for this purpose. Complete sets of these tools can be obtained from the Scintilla Company at moderate cost.

Shipment and Storage.—Do not pack more than four magnetos in a box. This will make a shipping weight of about 85 pounds per box, and in most instances will mean that the shipment will receive more consideration and care while in transit, since it is not too heavy to be handled by one man. Furthermore, the weight is such that the falls and jars incident to shipping are not so liable to cause damage. The box should be lined with moisture-proof paper. Fasten each magneto to the bottom of the box with two 3/8-16 cap screws. The length of the cap screws used will depend upon the thickness of bottom of box.

After being fastened in place, each magneto should be carefully wrapped in heavy wrapping paper and excelsior packed around it. The excelsior must be packed as solidly as possible by hand, using a sufficient amount to insure pressure upon it when lid is nailed down. Obviously, the tighter and more firmly the excelsior is packed, the safer it is for the magneto while in transit, in the event that any of them should become loosened from the bottom of the box. Place a steel band around the box. If more than one is used, space them so that they will afford maximum protection at the

mid-section. Print legibly or stencil the top of the box: "THIS SIDE UP." This tends to keep the magnetos in an upright position and does not impose too great a strain on bottom of box. Show consignor and consignee and how shipment is to be made. Clean each magneto thoroughly before storage. Apply vaseline to all exposed metal parts, wrap magnetos in paper and place on shelf in an upright position.

Type V-AG Scintilla Magneto.—The fundamental difference between the "V-AG" and the "AG" type magnetos lies in the method of mounting the large distributor gear as shown at Fig. 187. This assembly comprises basically, a new type distributor gear axle and a large distributor gear running on two open type ten millimeter ball bearings. Continuous lubrication from one major engine overhaul to the other is assured, thus eliminating the dangers experienced due to the lack of proper lubrication. The distributor gear axle is mounted in the front end plate in the usual manner. It extends through the hub of the distributor gear, having the gear and bearing assembly held in position on it by means of lock nuts. The adjustment of the bearing assembly is obtained by means of a spacer and adjusting washers that fit over the distributor gear axle and between the inside ball races. Grease is retained in the assembly by means of brass caps on each end of the distributor gear hub.

Disassembly of V-AG Distributor Gear Assembly.—Remove the brass cap from the end of the large distributor gear hub. Unscrew the lock nuts on the end of the distributor gear axle and lift the large distributor gear off. Note that when the large distributor gear is lifted out of position it carries with it the washer under the lock nuts, both of the cage and ball assemblies and the spacer and adjusting washers that maintain the bearing and play adjustment. Observe that when disassembled, the cage and ball assembly for the bearing in the distributor cylinder end of the large distributor gear hub, is a loose part whereas this assembly for the bearing in the other end of the gear hub is held in place by the brass cap which covers the bearing. Care must be taken that the same spacer and adjusting washers are used when re-assembling the bearing. The wear in this bearing assembly is practically negligible if properly assembled, hence the necessity of making another end play adjustment can usually be avoided by using the original spacer and adjusting washers. If the parts are not to be re-assembled at once, they should be oiled to prevent rust.

Inspection and Assembly of the Distributor Gear Ball Bearing Assembly.—Clean oil lead in front end plate thoroughly by means of compressed air. When compressed air is not available, flush the oil lead with light oil. The oil tube to the distributor gear axle is not used, due to the fact that the distributor gear ball bearing assembly is packed with nonfluid oil (fibrous grease). See that the fastening screws for the distributor gear axle are tight and locked in place. Note condition of balls and ball races. They run under very close tolerances and must be clean. Normally the balls and races will have highly polished surfaces and show no apparent signs of wear. Should any unusual wear or rough surfaces be noted, the parts so affected must be replaced with new ones. Examine the large distributor gear to see that there are no burrs on teeth of gear. Test dog screw. It must be tight and locked to the large distributor gear. Handle

the gear carefully as any strains set up as a result of improper handling will cause it to wabble, thus resulting in unnecessary wear of the teeth.

The adjustment of the bearing assembly can best be obtained when the parts are dry or have oil only on them. The assembly is accomplished as follows:—Place the large distributor gear on over the distributor gear axle the size of which is such that the inside ball races of the ball bearings have a slip fit over it. Now place the bearing spacer on the axle and then

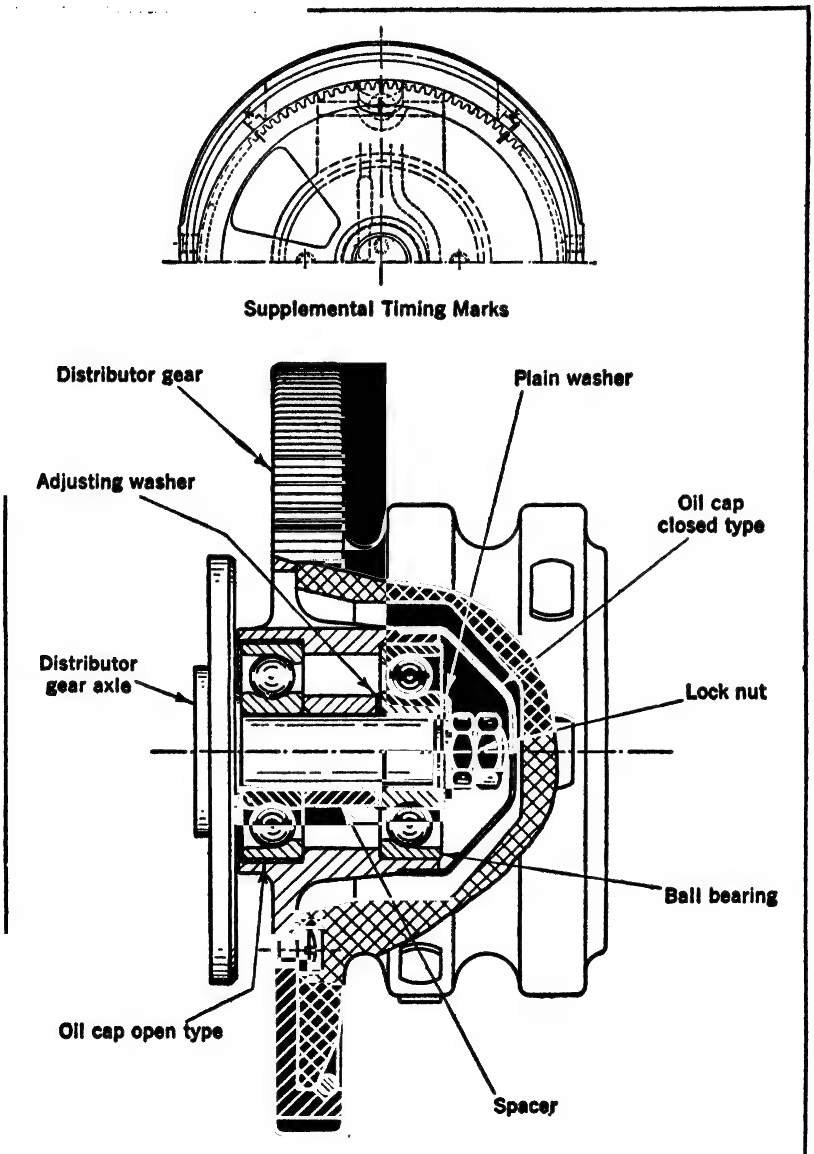


Fig. 187.—Ball Bearing Distributor Gear Assembly of Type VAG Scintilla Magneto.

the adjusting washers used in the original assembly. The cage and ball assembly for the bearing in the distributor cylinder end of the distributor gear hub should now be placed in position on the end of the distributor gear axle. Place the plain washer over the end of the distributor gear axle and on the inner ball race, after which the whole assembly should be drawn up tight by means of the lock nuts and the bearing adjustment checked.

Observe that the edge of the inside ball race extends beyond the shoulder on the threaded end of the axle, thus permitting the lock nuts to hold the inside races, the bearing spacer and adjusting washers, and the plain washer under them, as a stationary unit with the distributor gear axle.

Testing Bearing for Proper Adjustment.—Hold the front end plate with the distributor gear bearing assembly in both hands, having the bearing uppermost and the small end of the front end plate toward the operator. The adjustment of the bearing assembly for end play is checked by exerting a downward pressure on the rim of the gear with each thumb alternately and then moving the gear a few degrees at a time after each check until it has been turned through one complete revolution. The relatively high ratio of the large distributor gear radius to the radius of the ball bearing permits of obtaining a very accurate adjustment of this assembly. When properly adjusted there will be no end play in the bearing assembly and the gear will turn free when revolved by hand. If end play is found in the bearing it may be corrected by removing a thin steel adjusting washer. Should the removal of the washer make the bearing too tight a thinner one should be used to replace the washer previously removed. In the event that the adjustment desired is less than can be obtained with the adjusting washers on hand, recourse may be had to lapping one end of the bearing spacer. Adjusting the bearing by this method requires that the spacer be lapped carefully on one end only. A micrometer must be used to insure that the overall length of the spacer is uniform at all points on the circumference.

Assembly of Distributor Gear Bearing Assembly.—Having correctly adjusted the bearing, it must now be disassembled and packed with a good grade of nonfluid oil (fibrous grease). The hub of the gear should be packed with the above mentioned nonfluid oil after which the gear should be placed on the distributor gear axle and the bearing re-assembled. Draw the lock nuts up tight. Pack a small amount of the nonfluid oil in the brass cap that fits over the end of the gear hub and press it on in place. Replace the distributor cylinder and lock it in position by means of the steel spring ring.

Scintilla Magneto Types.—The Scintilla line includes a variety of magnetos and types are available for any type of engine. The AP5D is for five-cylinder radial engines and is driven at 1¼ times crankshaft speed. The GN6D is for six-cylinder engines and is driven at 1½ times crankshaft speed. The MN7D is for seven-cylinder radial engines and is driven at ⅞ times engine speed. These three types each weigh approximately twelve pounds. The PN4D is for four-cylinder engines. On the conventional type of motor it is driven at engine speed but on the "Cam" type

it is turned at twice engine speed. The AP3 for three-cylinder engines is driven at 1½ times engine speed. The MG2 is for two-cylinder engines and is driven one or one-half times the crankshaft speed depending on engine type.

The Scintilla Vertical Double Aircraft Magneto, Type SC is shown at Fig. 188 C. This magneto has one rotating magnet producing two independent sparks every 90 degrees of its rotation. The sparks may be synchronized or staggered as required by a simple adjustment of the breakers. The magnet is rotated between two pairs of pole shoes which causes simultaneous reversals of magnetic flux through the cores of two coils. The primary circuit of each coil is interrupted by one breaker mechanism of the spring type, the contact points of which are

Fig. 188.—Typical Forms of Scintilla Aircraft Magneto.

opened and closed through the medium of an extremely soft four lobe cam mounted directly on the end of the four pole rotating magnet. This magneto lends itself readily to radio shielding. The weight of the magneto alone is only 12.5 pounds. The two distributor heads to be mounted on a twelve-cylinder engine weigh 1.5 pounds each. By changing the drive gear ratio this magneto is adaptable to four-, five-, six-, seven-, eight-, nine-, ten-, twelve-, fourteen-, sixteen- and eighteen-cylinder engines when the required distributors are used. Simplicity, compactness and light weight are the chief characteristics of this magneto.

Instructions for the Use of Scintilla Aircraft Magneto Service Tool Set 4-273

23148—Tool for removing spring ring from distributor gear axle. This tool has two wedge-shaped blades so spaced that they fit, one on either side of the distributor gear axle: and two jaws which fit in between the open ends of the spring ring. The spring ring is removed by spreading it open with the jaws, at the same time forcing upward with the tool.

To Operate.—Push ends of wedge-shaped blades in space between spacing washers on end of large distributor gear axle bearing and spring ring. Insert the jaws between open ends of the spring ring. Spreading the jaws by pressing their opposite ends together with thumb and forefinger, and at the same time lifting the tool upward forces the spring ring out of its groove.

23123—Cam wrench for breaker cam. This tool is used to hold cam when tightening or removing cam fastening screw.

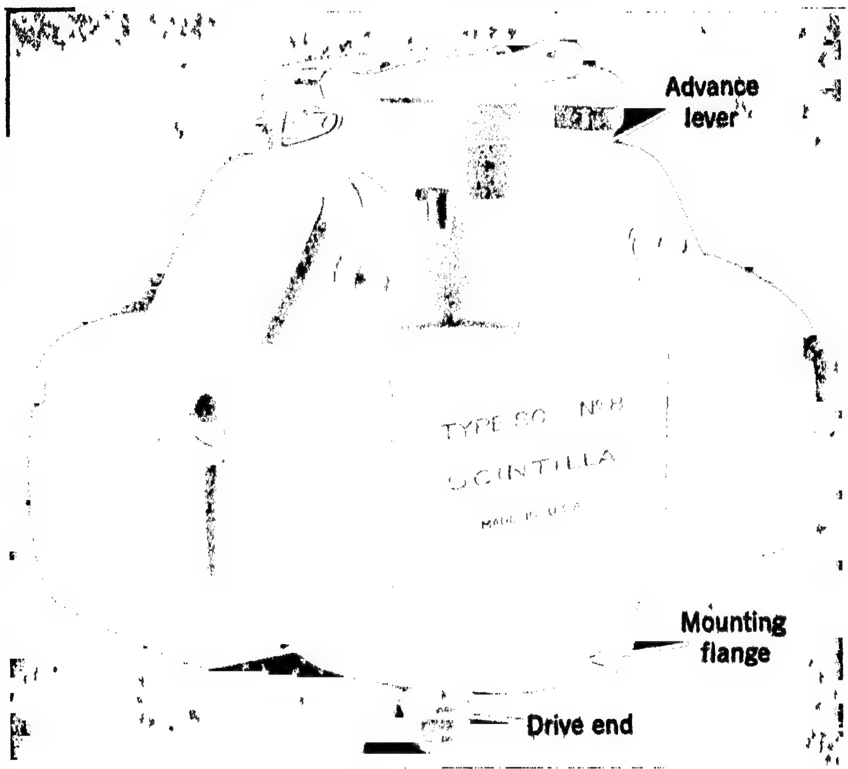


Fig. 188C.—Scintilla Type S. C. Double Magneto.

4-147—End wrench for lock nut on front end plate.

4-192—Tool for tightening electrodes in distributor blocks

20935—Socket wrench for spring clamp This wrench has an opening milled on the side of the socket, thus permitting it to fit over spring clamp and on spring clamp stud.

4-168—Socket wrench for drive shaft nut.

22033—Set of eight screwdrivers These screwdrivers are properly ground to prevent damage to slots in screw heads.

4-141—Template for electrodes in distributor block (twelve cylinder).

4-140—Template for electrodes in distributor block (nine cylinder)

4-139—Template for electrodes in distributor block (eight cylinder).

These templates will fit the distributor block in only the correct position. The radius of curved part of each template is such that it includes

the manufacturing tolerances and necessary clearance. Therefore, each template should fit its respective type of distributor block and not be held off by the electrodes touching curved part of template, i.e., the ends of the electrodes should just clear the template as it is held in position against the distributor block and moved back and forth over them.

4-271—Magneto-meter. Place the laminated end of the rotor in the Vee-shaped block extending from the case. Be sure that the magnet touches over the entire length of the block. Compare the reading of the meter with figures given on the card pasted to the inside of the instrument cover. To insure a satisfactory operation of the magneto, rotating magnets showing less remanence than the middle figure given must be magnetized.

23088—Mounting tool for inside ball race—seventeen millimeters. This tool is bored out to a depth that will permit of its being used for mounting either the small distributor gear or the inner ball race for the drive end bearing. The open end of the tool is counter-bored, thus permitting it to fit over the inner ball race and bear evenly against it while the race is being driven on.

To Install Small Distributor Gear.—Put a few drops of oil on drive shaft then place gear over shaft and bring it up to starting position. Note that the keyway in gear is in line with the Woodruff key in drive shaft. Place tool over drive shaft and hold its open end squarely and firmly against face of gear. Note that gear is starting on shaft evenly. Drive gear on until it stops against shoulder provided for it on the magnet. Best results are obtained by grasping the rotating magnet in such manner that while the magnet is held by the hand, the thumb and forefinger are guiding and supporting the tool. Hold the arm against the body in such a position that the impact from driving the gear in place will be absorbed by the body. Do not attempt to use a press for this work.

To Install Inner Race.—Seventeen millimeters. As previously stated, tool No. 23088 will be used and in general the instructions as given above for installation of the small distributor gear are to be observed.

23087—Mounting tool for inside ball race—fifteen millimeters. This tool has a counterbore in the open end that will permit it to fit over the inner race, thus holding the tool evenly against the inner race while it is being driven on.

To Install Race.—Put a few drops of oil on race seat on rear end shaft. Take the ball race and place it over the rear end shaft and on the edge of race seat. It is imperative that the ball race be placed on edge of race seat squarely, thus avoiding considerable difficulty and possibility of damaging the ball race seat.

Place tool over ball race and drive it on until back edge of race seats firmly against shoulder provided for it. The race should have a light drive fit on its seat. Best results are obtained by grasping the rotating magnet in such a manner that while the magnet is held by the hand, the thumb and forefinger are guiding and supporting the tool. Hold the arm against the body in such position that the impact from driving the gear in place will be absorbed by the body. Do not attempt to use a press for this work.

22175—Mounting tool for spring ring on distributor gear axle. This tool is composed of two parts. The inner part is used to expand the spring

ring while the tool body slides the spring ring into a position that will permit it to snap into place in the annular groove provided for it in the distributor gear axle.

To Operate.—Place spring ring on tapered surface of inner part of tool, then put tool body on over inner part of tool until the edges of open end of tool body rest upon the spring ring. Place the large end of inner part of tool on end of distributor gear axle shaft and press tool body down over taper, thus forcing the spring ring down and expanding it. As it leaves the large end of taper, the ring will snap into place in its groove on end of distributor gear axle.

20608—Universal gear puller. This puller is of the conventional double arm type. The puller bar has four holes for the puller arm fastening screws, thus providing for necessary adjustment for gears of different diameter. It will be observed that the two outside holes are to be used when pulling the eight cylinder small distributor gear.

To Operate.—Assume that the gear to be pulled is a nine cylinder small distributor gear. The puller arm fastening screws will be in the two inside holes in puller bar as previously stated. Place the brass cap over threaded end of drive shaft. Turn screw handle in an anti-clockwise direction until the inside end of screw will clear end of brass tip on drive shaft as the puller arm jaws are being placed in position behind gear. Place puller arm jaws behind gear, holding them in position by placing one hand under and around them. Lock the puller arm jaws in position by turning screw handle in a clockwise direction until inside end of screw bears firmly against end of brass tip on drive shaft.

To Remove Gear.—Continue turning the puller screw handle in a clockwise direction.

21281—Socket wrench for cam fastening screw and ground contact stud.

4-181—Socket wrench for booster current electrode.

MA-1635—Contact screw wrenches.

4-136—Mounting tool for outside ball race—seventeen millimeters. This tool has a No. 1 Morse taper which permits of its being used in either a drill press or a lathe. When used on a drill press the tool is mounted in a vertical position. The front end plate in which the seventeen millimeter outer race is to be installed is held horizontally with the flat boss on outside of front end plate resting squarely on drill press table. When used on a lathe, the tool is mounted in the tail stock. The front end plate is held against the lathe face plate. The diameter of the face plate must not exceed seven inches as a larger face plate would allow the oil cover axle bosses to rest on its edge, resulting in mis-alignment and serious difficulty in installing the race.

When installing the race by either of the foregoing methods, special attention must be given that the correct alignment is obtained. Note that the recess in paper washer which goes back of race is registering with the milled recess in the front end plate. Spread a few drops of oil evenly on one side of the insulating strip and place it in position with the oiled side inside. Note that the ends of paper strip are overlapping in recess milled for them in the side of race container in front end plate. Place ball race in position. Note that it is level and starts evenly when pressure is applied. The race

should have a light press fit in its container. After race is in place, the narrow width of the insulating strip extending above race must be cut off with a sharp instrument.

4-134—Mounting tool for outside ball race—fifteen millimeters (with supports for bearing part of magneto housing). This tool has a No. 1 Morse taper which permits of its being used in either a drill press or a lathe. There is also incorporated in this tool a device for holding the ball race in place on the end of the tool while it is being inserted in the magneto housing. A special pressing block is also provided which fits into the breaker compartment of magneto housing and rests on the drill press table or lathe face plate, as the case may be.

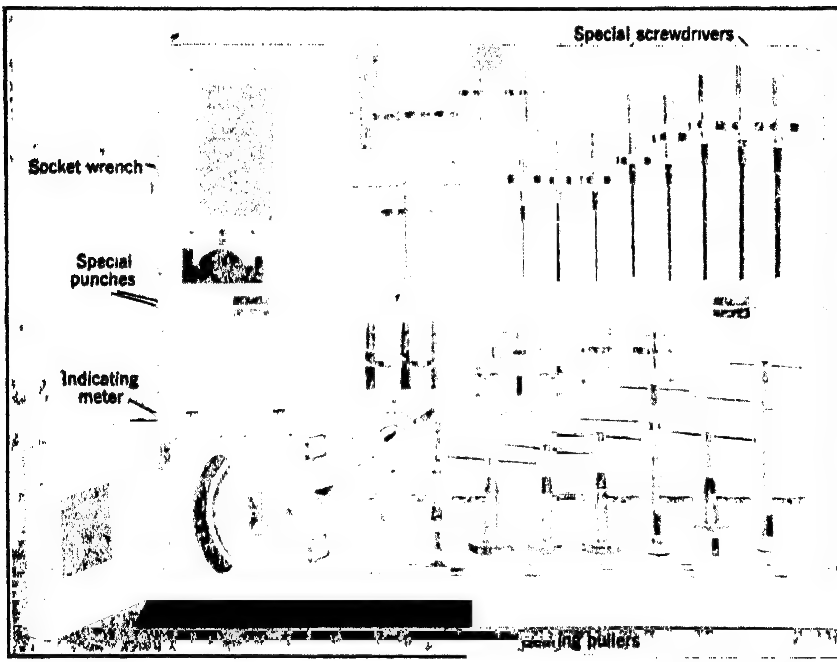


Fig. 189.—Special Tool Kit for the Inspection and Repair of Scintilla Aircraft Magnetos.

To Install Race.—Note that the paper washer which goes back of race is in place. Spread a few drops of oil evenly on one side of the insulating strip and place it in position with the oiled side inside. Note that the ends of paper strip are overlapping in recess milled for them in the side or race container in magneto housing. Place the ball race in position on end of tool. Have the pressing block in place in breaker compartment of magneto housing and place them so that the back surface of block will either rest on drill press table or against lathe face plate, as the case may be. Insert tool with ball race in magneto housing. Correct alignment of ball race with container may be best observed through opening between the pole shoe extensions on the magneto housing. The race should have a light press fit. After it is in place, the narrow strip of the insulating strip extending above the race must be cut off with a sharp instrument.

21203—Puller for small twelve cylinder distributor gear.

To Operate.—Turn screw handle in an anti-clockwise direction until shaft extends at least 2½ inches beyond threaded end of collet. Slide lock-ring back until it is off tapered surface of collet jaws. Place tool over drive shaft and back over small distributor gear until pulling edges inside the collet jaws fit in behind the gear. Lock collet jaws in place around gear by sliding lock-ring forward on their tapered surfaces. It is imperative that the edge inside collet jaws be behind back edge of gear before locking collet around it, thus preventing damage to gear.

To Remove Gear.—Turn screw handle in a clockwise direction.

21202—Puller for inside ball race—seventeen millimeters.

To Operate.—Turn screw handle in an anti-clockwise direction until shaft extends at least 3¾ inches beyond threaded end of collet. Slide lock-ring back until it is off tapered surface of collet jaws. Push tool on over ball race and back until open end of collet bears against face of small distributor gear. Lock collet by sliding lock-ring forward on tapered surface of collet jaws until it seats firmly. It is imperative that the pulling edges of the collet jaws be locked behind and around the back edge of this ball race on all types of Scintilla Aircraft magnetos.

To Remove Race.—Turn screw handle in a clockwise direction.

20709—Puller for inside ball race—fifteen millimeters.

To Operate.—Turn screw handle in an anti-clockwise direction until shaft extends at least 2½ inches beyond threaded end of collet. Slide lock-ring back until it is off tapered surface of collet jaws. Push tool on over ball race and note that the collet jaws are bearing on the annular groove of ball race. Lock tool to ball race by sliding lock-ring forward on tapered surface of collet jaws until it seats firmly.

To Remove Race.—Turn screw handle in a clockwise direction.

22972—Puller for outside ball race—seventeen millimeters. Hold body of tool in a vertical position and turn nut on threaded shaft in an anti-clockwise direction until the shaft has dropped through tool body sufficiently that the expanding collet on lower end of shaft is at least ½ inch below the lower end of tool body. Turning the "T" handle on end of threaded shaft in a clockwise direction lower taper in expanding collet, thus permitting its spring tension to close collet so that it may be inserted through bearing race until it rests on back wall of race container. Turning the "T" handle on end of threaded shaft in an anti-clockwise direction draws the taper on lower end of shaft upward, thus expanding and locking the wedge-shaped jaws of collet behind the ball race. Turning nut on threaded shaft in a clockwise direction causes lower end of tool body to bear against front end plate and exerts a considerable pull on threaded shaft, thus removing the outer bearing from its container in the front end plate.

20767—Puller for breaker cam.

To Operate.—Turn screw handle in an anti-clockwise direction until end of screw is at least two inches from threaded end of collet. Slide locking sleeve off tapered surface of collet jaws. Place open end of collet over cam and move tool back until the raised surface inside collet jaw is behind ground surface of cam and over its small diameter. Slide locking sleeve forward on tapered surface of collet jaws, thus locking them in place around

cam.

To Remove Cam.—Turn screw handle in a clockwise direction.

22973—Puller for outside ball race—fifteen millimeters. Hold body of tool in a vertical position and turn nut on threaded shaft in an anti-clockwise direction until the shaft has dropped through the tool body sufficiently that the expanding collet on lower end of shaft is at least ½ inch below the lower end of tool body. Turning the "T" handle on end of threaded shaft in a clockwise direction lowers taper in expanding collet, thus permitting its spring tension to close collet so that it may be inserted through bearing race until it rests on back wall of race container. Turning the "T" handle on end of threaded shaft in an anti-clockwise direction draws the taper on lower end of shaft upward, thus expanding and locking the wedge-shaped jaws of collet behind the ball race. Turning nut on threaded shaft in a clockwise direction causes lower end of tool body to bear against magneto housing and exerts considerable pull on threaded shaft, thus removing the outer bearing race from its container in the magneto housing.

4-148—Keeper for pole shoe extensions. This tool is placed across pole shoe extensions to avoid partial demagnetizing of rotating magnet when it is turned without coil being in place.

4-145—Rotor handle. This handle screws on the drive shaft and is useful when rotating magneto by hand for checking internal timing, etc.

4-149—Die 3.5 millimeters Lowenherz for contact screw.

4-150—Tap 3.5 millimeters Lowenherz for support and lever.

Tap and die for use in cleaning up contact screw threads.

4-191—Gauge for safety gap.

4-146—Feeler gauge for setting contact points.

20906—Reamer for breaker lever bushing. This reamer is tapered on more than one-half its cutting length, hence the correct size for breaker lever axle bushing is not obtained unless the reamer is run through the bushing its full length.

22683—Mounting tool for oil cover axle.

22684—Mounting tool for spring clamp axle.

These tools are magnetized so that they will hold their respective axles.

22682—Three drifts for axles 1.5 millimeters.

22682-1 .... 1.8 x 35 millimeters.
22682-2 .... 1.8 x 20 millimeters.
22682-3 .... 1.8 x 10 millimeters.

22681—Three drifts for axles three millimeters.

22681-1 .... 2.8 x 32 millimeters.
22681-2 .... 2.8 x 15 millimeters.
22681-3 .... 2.8 x 45 millimeters.

22045—Locking tools for divers screws.

22045-1 .... 0.7 millimeters.
22045-2 .... 0.5 millimeters.
22045-3 .... 0.3 millimeters.

CHAPTER XV

SPLITDORF AIRCRAFT MAGNETO—BATTERY IGNITION SYSTEM

Splitdorf Aircraft Magnetos—Splitdorf "S" and "SS" Magnetos—Mechanical Operation—Electrical Operation—Inspection and Testing Model "SS" Magneto—Splitdorf Double Magneto—VA Magneto Details—Limitations of Lever Type Breaker—Pivotless Breaker a New Development—Delco Battery Ignition System—Delco Wiring Diagrams—Characteristics of Ideal Ignition System—Radio Shielding—Sparkplug Design and Application—Two Spark Ignition—Electric Generators for Airplanes—Third Brush Regulation—Voltage Regulated Generators—Aircraft Storage Batteries.

Splitdorf Aircraft Magnetos.—The Splitdorf Electrical Co. of Newark, N. J., has recently brought out two new magnetos for aircraft engines which are remarkable on account of their low weights. One is the NS9, for the ignition of nine-cylinder radial engines, which by changing certain parts and running it at lower speeds relative to the engine crankshaft can be used also for engines with a lower number of cylinders, down to three; the other is the VA2, a double magneto, which generates twice as many sparks per revolution of its rotor as the NS model, and can be used to provide double spark ignition for aircraft engines of from three to twelve cylinders.

Both magnetos are of the inductor type and embody the peculiar magnetic field structure which has been characteristic of Splitdorf magnetos for many years. The primary and secondary windings are wound on a core of U shape, which is provided with separate pole pieces having polar surfaces conforming to the poles of the rotor. The rotor of the single magneto has two poles at each end, those at one end being at 90 degrees to those at the other end. At right angles to the inductor poles in the rotor tunnel are the poles of the permanent magnet. The poles of the rotor therefore, form magnetic bridges which carry the flux from the poles of the magnet to the coil core. When a rotor pole spans both a magnet pole and an inductor pole the magnetic circuit is completed, and when the trailing edge of the rotor pole passes out from under the magnet pole or the inductor pole the magnetic circuit is interrupted. It is, of course, this opening and closing of the magnetic circuit through the coil core that results in the generation of electric impulses in the high tension coils.

The present type of single magneto for nine-cylinder engines (the NS9) weighs not quite ten pounds, but by the substitution of cobalt for tungsten steel magnets it is hoped to reduce the weight of the complete magneto to eight pounds. This is shown at Fig. 190A.

All of the cast parts of the magneto are aluminum alloy die castings, the four more important ones being the rotor, the magneto housing and the front and back plates. The breaker housing also is a die casting, while the breaker base is a bronze forging. All parts of the magnetic circuit except the magnetos themselves are made of transformer stock (silicon iron) 0.008 inches thick. There are four different assemblies of these laminations,

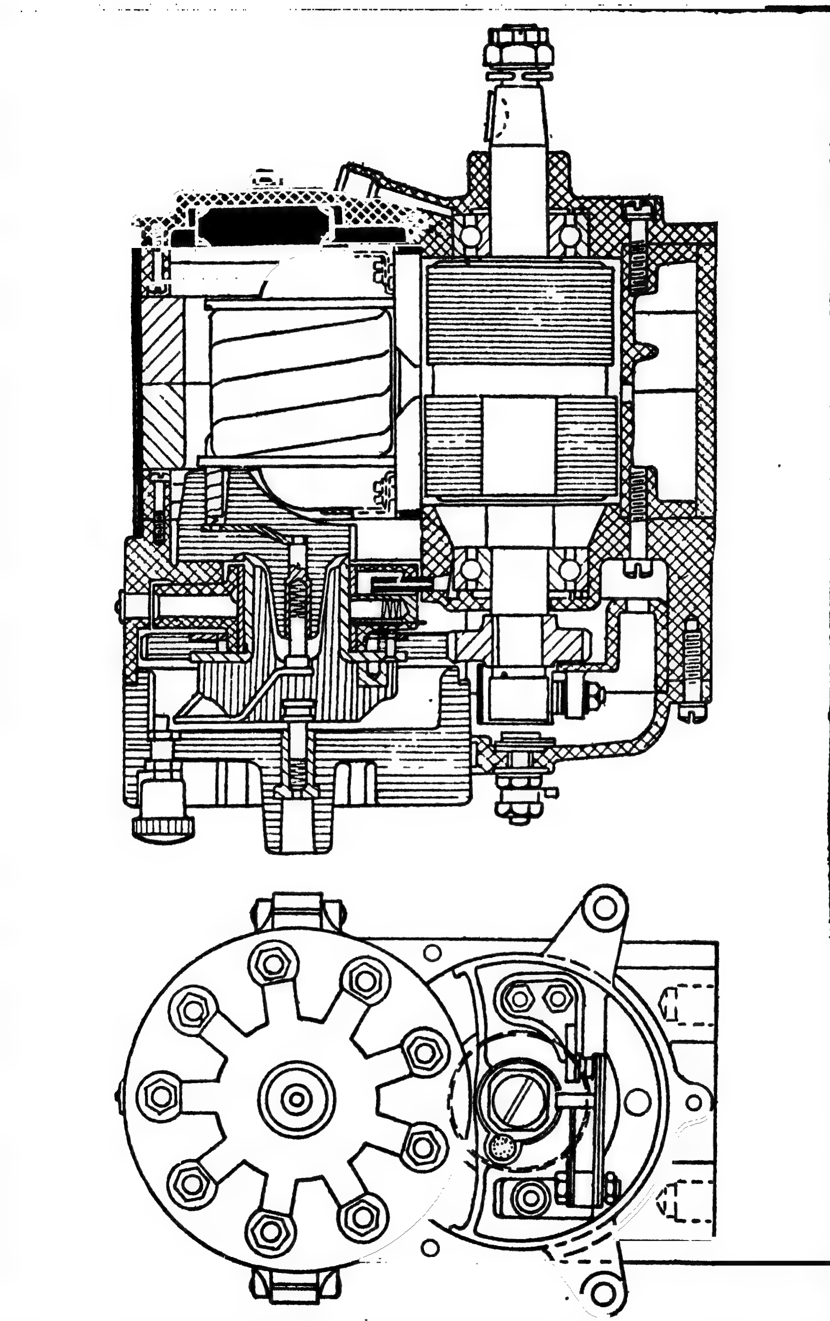


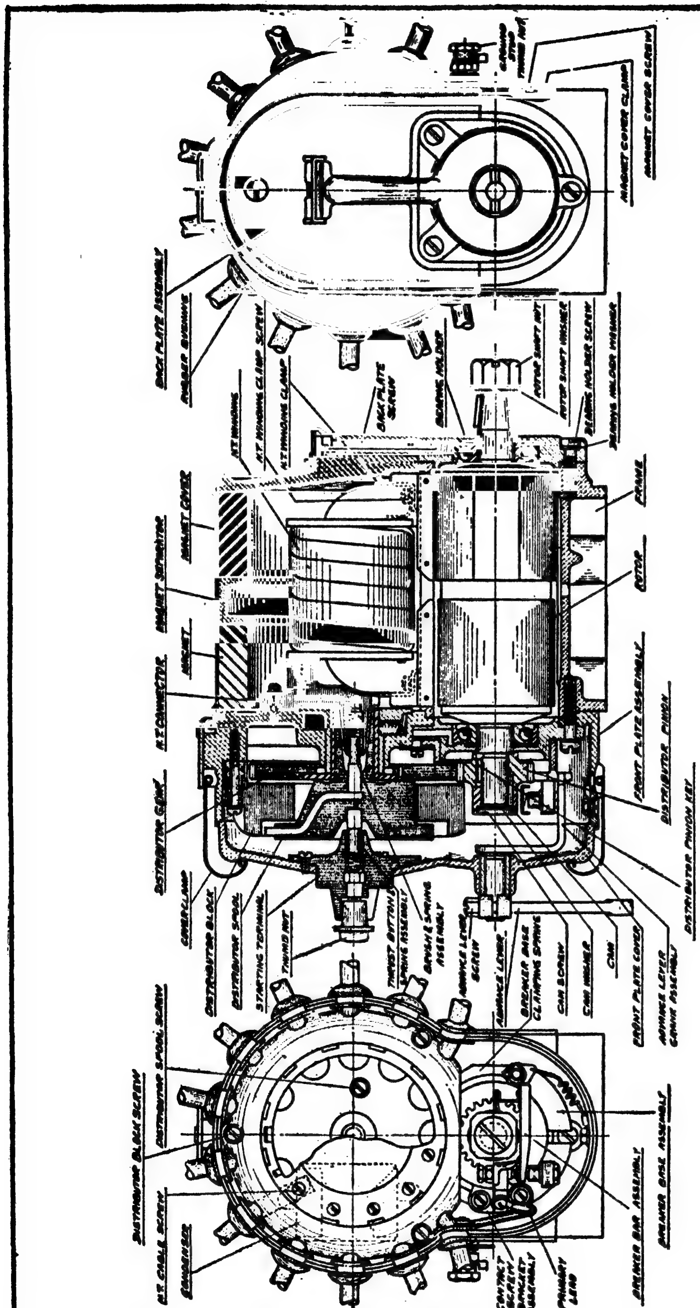
Fig. 190A.—Front Elevation and Longitudinal Section of Splitdorf NS-9 Inductor Type Magneto.

viz., the magnet poles, the rotor poles, the inductor core and the inductor poles. The rotor, which weighs one and one-half pounds, has a shaft made of three and one-half per cent nickel steel, heat treated and ground, and is supported in Norma Hoffman deep groove ball bearings.

The coil core, after being insulated, is wound with primary winding at the bottom and with the secondary winding over the primary. The secondary winding consists of enameled copper wire 0.002 inches thick, and sheets of paper and varnished silk are placed between the layers of wire. After the coils are completely wound they are impregnated with insulating varnish in a vacuum and then subjected to pressure in order to force the insulating compound into coils. This process comprises a number of steps. The coils are first heated and are then placed in a cast-iron cylinder from which the air is exhausted by means of a suitable vacuum pump. By opening a valve, an insulating compound is then allowed to flow from an adjacent container into the vacuum chamber, until it completely covers the coils therein. Next, by opening another valve in a pipe from a compressed air tank to the top of the vacuum chamber, air is admitted to the latter until a pressure of 90 pounds per square inch is exerted on top of the insulating compound, which latter is forced into all of the pores of the coil. The air pressure is left on for half an hour. The coils are then removed from the vacuum chamber and baked in electrically heated ovens for 24 hours, allowed to cool, and baked a second time.

The breaker used on these magnetos is of the high speed type and is operated by a four-lobe cam on the rotor shaft. The breaker arm itself is of spring material. Between it and the cam there is a bumper of hard fiber which is held at the end of a laminated spring, being riveted to the end of the main leaf, which is bent at right angles. The breaker points are either of precious metal (a platinum-iridium alloy) or of pure tungsten. They are adjusted to open 0.008 inches by means of a pinion and rack mechanism which moves the so-called movable breaker point in the direction of motion of the fixed point. These details are best shown in one of the end views of the double magnetos.

The chief object of the double magneto shown at Fig. 191 is to produce two simultaneous or nearly simultaneous sparks in the combustion-chamber of each cylinder. In many of the cases one of the sparkplugs is located adjacent to or opposite the exhaust valve, while the other is similarly placed with relation to the inlet valve. It has been found that where such double ignition is employed the best results are obtained if the spark occurring near the exhaust valve is given a slightly greater advance. This is undoubtedly due to the fact that the flame travels slower in a mixture containing a somewhat greater proportion of spent gases, as does the portion of the charge near the exhaust valve. In order to make it possible to time the two sparks independently, the breaker base is made in the form of two semicircular discs. With the double magneto two breaker arms are used which would normally be placed exactly opposite each other. However, in order to permit "staggering" the sparks by a few degrees, the semicircular breaker bases are made with oblong holes through them, permitting one of them to be adjusted angularly with relation to the other.



Sectional View SS-12 Aircraft Magneto

Fig. 190B.—Front and Rear Elevation and Longitudinal Sectional View of Type SS-12 Splitdorf Magneto.

The double magneto is designed for vertical mounting, which has been found of advantage from several points of view. It makes the drive very simple, as the magneto can be driven directly from the crankshaft through a pair of bevel gears, and the mounting on a machined surface on the crankcase extension is also very convenient. One magnet is placed on each side of the rotor and its housing. The arrangement of the poles in the rotor tunnel is somewhat different from that in the single magneto. In the latter, as explained, the magneto poles are at right angles to the inductor poles, but in the double magneto the angular spacing is 60 degrees, it being necessary to accommodate four magneto poles instead of two. Aside from this difference, the magnetic circuit of the double magneto is quite similar to that of the single machine. The condensers used with these magnetos are built of alternate layers of tinfoil and mica and have a capacity of one-quarter micro-farad each. In the single magneto the condenser is placed in a pocket formed in the back end plate, while in the double magneto the two condensers are placed similarly in pockets formed in the top plate, on opposite sides of the interrupter housing.

With the latest design the distributors are separate parts and mounted on the end of the engine camshafts. There is a single high-tension connection between the magneto and the distributor. In the case of the double magneto there is a single high-tension cable from each coil to each distributor. These high-tension cables are completely shielded to prevent interference with radio reception. At the outlet from the magneto there is an L-type fitting of die-cast aluminum, and from this to the distributor the cables are protected by tinned copper braiding. Thus all parts carrying the high-tension impulses are covered by metal and the emission of waves from them is prevented.

Like all magnetos, the Splitdorf is provided with a safety spark gap which prevents the voltage in the secondary from building up to dangerous values when there is an unusually high resistance in the secondary circuit, as when the sparkplug cable is disconnected. In the single magneto illustrated herewith the safety spark gap is shown to be between the service brush and a grounded spark point located on the distributor gear. With the double magneto there are two such safety spark gaps, each protecting one of the high-tension windings. Owing to the fact that if there is repeated sparking at the safety gap in an enclosed space, the air within this space becomes ionized and then loses much of its electrical resistivity, permitting the spark to pass at the safety gap at much lower voltage than normally, it is necessary to ventilate the inside of the magneto.

There is some objection to having the safety spark gap exposed on the inside of the magneto, for the reason that the sparks produce nitrous oxide which attacks the cuprous materials, forming copper nitrate. In the latest design, therefore, the safety gap is located inside a small porcelain tube which is immune from attack by the nitrous oxide. However, owing to the reduced space within the tube the air therein would become ionized very quickly if sparks should pass and effective ventilation were not provided. This ventilation, however, must be effected in such a way that there is no possibility of the sparks at the safety gap igniting any combustible mixture that may be present in the vicinity of the magneto. This is accomplished

by ventilating the chamber through a long, small-diameter hole in the metal housing of the magneto, which will quench any flame that may start inside the spark gap housing before it reaches the atmosphere.

The distributors used with these magnetos may also distribute the spark from an auxiliary starting magneto. This spark enters the distributor by way of a bronze ring moulded in the housing and is picked off therefrom by a radial rotating electrode, or "brush." This latter connects with another rotating electrode, the point of which sweeps past the distributor sectors. The starter "brush" trails behind the service "brush" in such a way that the starting spark occurs in the cylinder when the piston in it is on the down stroke, so that the engine cannot possibly start in the reverse direction.

The Splitdorf S and SS Magnetos.—The design of the Splitdorf models S and SS line of magnetos is based on years of research and construction of both armature and inductor type instruments. They incorporate the mechanical and electrical improvements necessary to make them rugged in construction, simple in design, enclosed dust, oil and water-proof, with such spark intensity to spark at extreme low speeds and under high compression. This magneto is of the inductor type as shown at Fig. 190 B and the design is based on years of research and construction of both armature and inductor type magnetos. It incorporates the mechanical and electrical improvements necessary to make it the last word in an ignition instrument. In the following semi-technical description the salient features will be explained under two captions, electrical and mechanical. This is done for the sake of simplicity and the points will be noted in their proper relative order proceeding from both the mechanical and electrical sources of energy.

Mechanical Operation.—Starting at the drive end, which is of standard S. A. E. dimensions, the steel rotor shaft is of one piece passing entirely through the rotor. It carries the rotor inductors and is supported on ball bearings. The laminated soft iron inductors are fastened to the shaft by die cast aluminum. The shaft is of smaller section in the center and is flattened and provided with a driving pin which insures a very substantial means of carrying the rotor inductors on the driving shaft while at the same time providing magnetic insulation. The two ball bearings on the shaft are carried in end plates which fasten to the frame of the magneto. These end plates are doweled into the bore of the frame, insuring perfect alignment. The frame is of die cast aluminum in which are set two magnet pole pieces and two coil pole pieces. It will be seen that the method of securing the inductors and the rotor and the pole pieces and the frame provides the greatest mechanical ruggedness possible. The breaker, cam and distributor pinion are carried on the rotor shaft, following out in general the customary Splitdorf design. The breaker mechanism and the distributor are completely enclosed in a compartment formed by the front plate and cover. The breaker mechanism is attached to a base carried in the frame and capable of sufficient movement to provide for advance and retard. The breaker bar is of a light weight pressed steel construction, grounded through a stranded copper strip to the frame of the magneto.

The model SS has its condenser attached to the front plate and is placed behind the distributor gear. The model S condenser is carried on two studs

attached to the breaker base, a flat spring fastened to the metallic condenser case prevents the breaker bar from coming off the stud but provides easy access to it and facilitates its removal. The model SS platinum contact points are of the conventional design and adjustment is made in the usual way by moving the stationary contact screw. The model S stationary contact point is carried on an adjustable bracket which provides the necessary adjustment for setting the point. It also has the added advantage that in adjusting the points the stationary point is not rotated, thereby insuring the two contacting surfaces always being parallel and retaining their original seating. The method of adjusting the point is described by a sketch and printed paragraph on the surface of the condenser. Thus each magneto carries its own instructions for adjusting the points.

The distributor gear which meshes with the steel driving pinion on the rotor shaft is carried on its steel shaft in a plain, well lubricated bearing. The distributor disc is supported by the distributor shaft and attached to it. The front cover which completely seals the breaker and distributor compartment carries the advance lever. The advance lever is attached to a shaft and by means of an arm to the breaker base. The design is sufficiently flexible to allow placing the advance lever in the right, left or vertical positions. The model SS distributor block is completely enclosed in the distributor and breaker compartment. The sparkplug cables which attach to it are carried out through the frame in rubber grommets. The model S front cover which completely seals the breaker and distributor compartments carries the distributor block and the advance lever. The advance lever is attached to a shaft and by means of an arm to the breaker base. The design is sufficiently flexible to allow placing the advance lever in the right, left or vertical positions. The moulded distributor block carries the distributor brushes and provides a novel, quick and water-proof means of attaching the sparkplug cables.

MODEL SS-12 AIRCRAFT MAGNETO

Specific No.	Rot.	Shaft Height	Dist Block Firing Angle	Gear Ratio	Driving Speed 4 Cycle	Remarks
SS-12-4313	RH	50mm	30°	3–1	1½–1	Aircraft
SS-12-4314	RH	50mm	30°	3–1	1½–1	Aircraft F S
SS-12-4315	LH	50mm	30°	3–1	1½–1	Aircraft
SS-12-4316	LH	50mm	30°	3–1	1½–1	Aircraft F. S.
SS-12-4317	LH	55mm	30°	3–1	1½–1	Wright-Aircraft
SS-12-4318	RH	50mm	30°	3–1	1½–1	Curtiss

Electrical Operation.—The horseshoe magnets (model S having one, model SS two) are secured to the magnet pole pieces of the frame by the magneto cover. The flux is taken from a comparatively small area at the ends of the magnets. The magnetic patch is through the magnet, the magnetic pole pieces, the rotor inductors, the coil pole pieces, and the coil core. The soft iron portion of this circuit is laminated. The rotor inductors are

properly spaced to generate current for sparks at the desired firing angle. These angles of the rotor inductors may be equal, as in all SS models, S4, etc., or they may be unequally spaced as in model S2-42 degree motorcycle type. This construction eliminates any possibility of the shaft becoming magnetized. The coil is wound on an inverted "U" shaped laminated core, the ends of which are enlarged to obtain sufficient area of contact with the coil pole pieces.

The primary and secondary coil connections are soldered to the coil terminals, eliminating screws at these points. The flexible insulated wire from the primary winding to the circuit breaker is carried through the breaker distributor compartment. The connection from the secondary winding to the distributor is carried through a well insulated connector which carries the high tension current to the stem of the distributor disc. The safety gap is located between the secondary winding of the coil and the distributor, where its electrical efficiency is greatest. The distributor disc is moulded from a newly developed material having a higher dielectric strength than other moulded insulators.

Inspection and Testing Model SS Magneto.—The following procedure is recommended by the Splitdorf Electrical Company for inspecting and testing the SS magneto in a systematic manner.—

1. See that the rotor turns freely. Be sure that you distinguish magnetic pull from a scraping or binding of the rotor.

2. See that all screws and nuts are tight, and have lock washers, cotter pins and seals where specified.

3. See that oil covers are not open excessively around the edge of oil hole. See that oil hole is open to the bearing.

4. Examine cam to see that it is not cracked or burred and that the surface is smooth. See that bumper rides squarely on cam.

5. See that primary connections are clean and make good electrical contact. See that the primary wire is well insulated and in its correct position. Primary wire should not have broken, unsoldered or unprotected strands.

6. See that the contact points line up and come together squarely; are free from oil or foreign matter and are properly set. The contact points should not be more than .004 inches out of line.

7. See that the bumper and contact points are securely riveted to the breaker arm. See that breaker bar ground wire is in the proper position.

8. See that the condenser contacts are clean and make good electrical connection.

9. See that the distributor gear and pinion have punch marks to show the proper meshing of the gears.

10. See that the primary wire clears the advance lever at bottom of breaker base plate in both the advance and retard positions.

11. The contact points must not open more than .024 inches nor less than .020 inches nor must they arc or flame excessively at any speed within the test range.

12. All brushes and thrust buttons must be so fitted that they may be completely pushed into their sockets and will freely return to their normal position. In normal position the brushes must stand perpendicular to the surface with which they contact. Do not change tension of carbon brush

springs. Very little tension is required.

13. The ends of carbon brushes must be square, smoothly finished and the edges not nicked.

14. The interior surface of the distributor block should be cleaned of any carbon deposits, also from distributor disc. With magnetos having adjustable brush jump spark or gap distributor, the clearance between segments in block and brush should be .010 inches.

15. The advance mechanism must operate freely and be held firmly in position by the clamping springs.

16. See that distributor block retaining wire holds the block tightly in place. (S models.) Clamping wire can be bent to suit.

17. The magnets on all magnetos should be put on with special keeper mentioned in tool list, and the polarity should be the same on all magnetos of the same specification number. North leg of magnets should be marked with red paint, extending down about one-quarter inch on base.

18. The gap between the rotor wing and pole piece in the full advance position when the contact points have separated .001 inches or when the buzzer ceases to vibrate, if tested with a buzzer should be from .035 inches to .050 inches inclusive, except the S2 42 degree and 45 degree in which the close break shall be set with $1\frac{3}{64}$-inch gauge which results in rotor break of approximately .050 inches between the pole and rotor wing. Fixed spark machines are to be locked in full advance position.

19. The magneto should be timed by turning the rotor in the proper direction of rotation and the contact points should remain closed until the distributor brush is fully on the segment of distributor disc or block in the full advance position.

20. Cam oiling wick should be saturated with heavy cylinder oil (600W) when placed in service. Ball bearings are to be packed with fiber grease or vaseline during assembly and no further oil or grease added before shipment.

Plain bearing machines are to be carefully oiled with a good grade of machine oil before road test and a few drops added to each bearing before shipment.

21. The spark should not jump the safety gap at any speed when each distributor wire is connected to an individual spark gap of $\frac{3}{16}$ inch.

22. Magnetos should cease to spark when the grounding terminal is short circuited.

23. Test speeds of aircraft magnetos—150 to 4,200 r.p.m.—full advance-sparking over five millimeter gaps of third point type in open air.

Splitdorf Double Magneto.—At the request of the Materiel Division of the Army Air Corps the Splitdorf Electrical Co., through its chief engineer, E. B. Nowosielski, developed the Splitdorf Model-VA vertical double magneto, as shown in Fig. 191 A in part section and Fig. 191 B as it appears ready for installation on an engine. It is interesting to note that the first experimental model was made by splitting two Dixie-100 hand-starting magnetos horizontally through the center line and bolting the two halves together with a common inductor-rotor. This type of magneto is of a very simple, compact and rugged design and is probably the first type specially designed for aircraft engines. Its features are:—

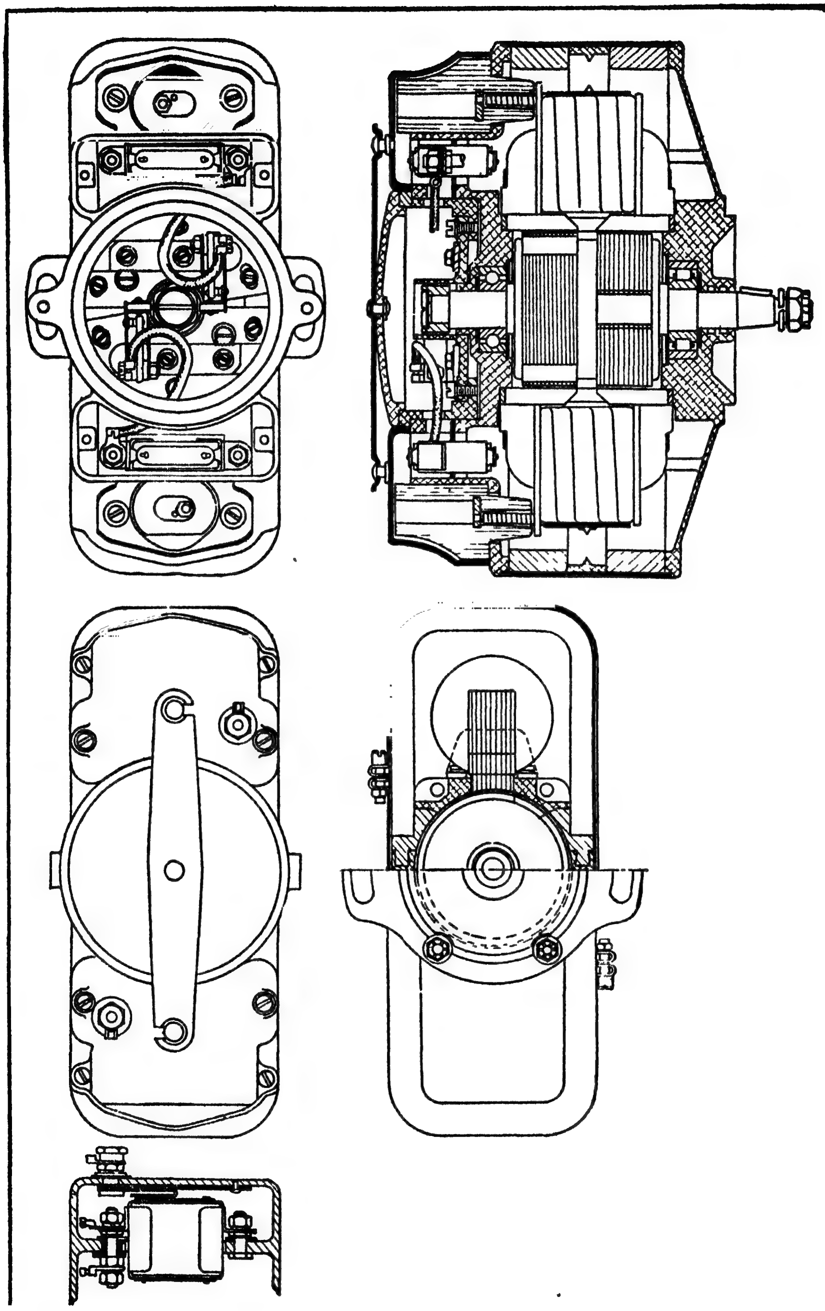


Fig. 191A.—Sectional and Outside Views of the Splitdorf VA-2 Magneto, for Aircraft Use.

(1) Two electrically independent sources of spark from one unit weighing less than one conventional single magneto
(2) Single shaft with flange mounting allowing direct splined shaft or gear drive and eliminating the necessity for special drive shafts and mounting brackets
(3) Constant angular relation between two sparks either synchronized or staggered; one spark-advance control-rod
(4) Distributors driven from the camshaft without special gear-reduction
(5) A single type of magneto for engines having any number of cylinders.

The construction of the VA vertical double magneto will be described in proper sequence.

Splitdorf VA Magneto Details.—The Splitdorf VA Aircraft Magneto is a most recent development. This instrument is made to be mounted vertically in the "Vee" of the engine, requiring but a single drive and one control rod to the magneto breakers. This instrument delivers two sparks simultaneously, or at spaced intervals. In effect the operation of this magneto is similar to two independent ignition units, whereas structurally it is a double magneto arranged into a single assembly with the parts requiring frequent inspections so placed that they are readily accessible and easily serviced. The Splitdorf VA Double Aircraft Magneto is of the inductor type, producing four double sparks per revolution. In its design many innovations in construction have been introduced with the object of securing a higher degree of electrical efficiency and a unit that would be mechanically stronger and constructionally more accessible with a material reduction in weight. It being possible to use aluminum alloys extensively in this design, together with the possibility to laminate in addition to the coil poles both magnet poles, coil cores and rotor inductors which convey the rapidly changing magnetic flux; there can be no doubt that these distinctive details account for the very efficient performance of these magnetos in service. By virtue of the laminated parts, eddy current dampening is reduced to a minimum, substantially improving the electrical characteristics of these instruments. The fat sparks produced by this type magneto are of greater intensity and duration than can be obtained from any other system of ignition.

This magneto consists of a frame, top plate and bottom plate, rotor, breaker housing, and magnets. Cast into the top and bottom portions of the frame on opposite sides are four separate coil poles, it being understood the other sides carry the magnet poles. The coil poles are bridged on each side by the laminated "U" shaped coil cores, the centers of which carry the usual primary and secondary windings commonly used in high tension magneto construction. The rotor consists of four laminated inductors spaced around the circumference and arranged apart from each other and displaced 90 degrees. The rotor shaft is of steel reduced at the center and die cast in place with the inductors, making a rigid and well balanced structure. The extended portions of the shaft carry the ball bearings, the cam on one end and the usual drive at the other end. The top and bottom plates are provided with suitable dowels. These register into the rotor bore and carry

ball races for the supporting of the magneto rotor. It will be noted that with the magnets in place, coils in position, and the rotor bridging the coil poles, causing magnetic lines of force to flow through the coil cores, an electric current is induced in the windings. With this disposition of poles, coils, inductors and magnets, it may be readily seen that one revolution of the rotor produces four complete flux reversals in each coil core, thus giving this instrument the capacity of producing four double sparks per revolution.

The "U" shaped magnets are made from tungsten steel, carefully hardened and accurately ground to give the largest possible surface contact with the pole pieces, producing a magnetic field of maximum strength and efficiency.

Briefly, the breaker is made up of two specially developed breaker bars provided with the usual contact points and springs and co-operating with the insulated adjustable contact screws. Provision is also made for the movement of one breaker plate a certain number of degrees so that one spark may be definitely timed from the other, permitting an earlier spark over the exhaust valves should this be desired. A single cam of conventional design actuates both breaker arms, giving four breaker openings per revolution. Cam lubrication is by oil wick. Two carefully made mica condensers of the proper capacity are mounted in the recessed portions of the top plate adjacent to the circuit breakers, shunted across the contact points, and connected to the primary windings in the usual manner. One of the advantages of this arrangement is the ease with which parts can be replaced when necessary.

The distributors are of the jump gap type, moulded from a rubber compound having high dielectric properties. The rather unusual method of mounting the distributor blocks on the camshaft housing has worked out entirely satisfactory. With this arrangement the distributor drive is of simple construction rarely requiring attention. The high tension leads from the coils terminate at the insulated connectors on each side of the magneto. Suitable cables convey the current to the distributor spools and from here it is distributed to the sparkplugs in the usual manner.

Limitations of Lever Type Breaker.—The conventional lever-type breaker mechanism of good design is satisfactory at sparking rates up to about 10,000 sparks per minute. Above this speed, the inertia of the breaker lever is usually sufficient to prevent it from following the movement of the cam, and chattering occurs. Rather than overcome this chattering by increasing the spring tension on the lever, with the resulting increased contact and rubbing-block pressures, a solid bumper, or a buffer-spring, is placed behind the lever to limit its travel and to allow the spring to return the lever to the closed position in time for another spark. This chattering greatly increases the wear of the contacts by introducing high impact-loads as the contacts close, and by causing additional arcing at the contacts when the lever bounces several times before coming to rest. So long as the lever follows the cam, the closing velocity is governed by the contour of the cam and can be made much less than is the case when chattering occurs.

A cylindrical pivot-bearing is usually provided for the contact-lever, either near the center or at one end. This bearing must have an insulating-bushing in the lever to prevent the pitting of the pivot-pin by electrical dis-

charges from the lever to the pin. These are usually high-frequency static discharges that occur in spite of the fact that the lever is grounded through the spring. The lubrication of this bearing is particularly difficult, as the motion is oscillatory, and foreign particles do not work out of the bearing; consequently, any rust, moisture or wear tends to cause the lever to stick. Some means for lubricating the pivot-bearing and cam must always be provided. Experience shows that such breakers require frequent attention when operated on high-speed engines.

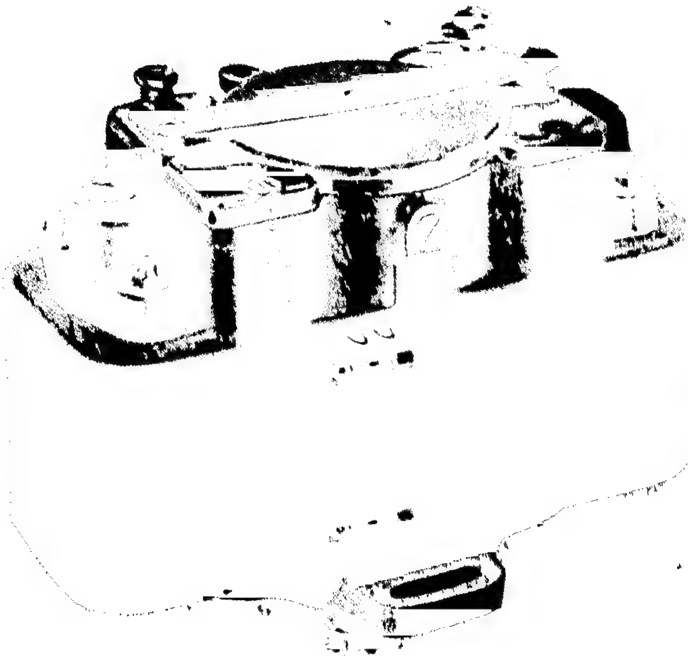


Fig. 191B.—The Splitdorf Model VA Double Magneto which Weighs but Thirteen Pounds.

Pivotless Breaker New Development.—In an attempt to eliminate the various troubles encountered in service with the lever-type breaker, the experimental engineering section of the Materiel Division undertook the development of a breaker to meet the following requirements:

(1) Ability to operate continuously at speeds equivalent to 3,000 r.p.m. on a twelve-cylinder engine, at 18,000 sparks per minute
(2) Minimum impact-loads on the rubbing-block and contacts
(3) Elimination of the pivot-bearing and lubrication of the breaker mechanism
(4) Simple adjustment of the breaker without special tools and without disturbing the alignment or refacing the contacts
(5) Elimination of flexible lead-wires from the primary winding to the breaker.

After the design and construction of a number of experimental breakers and several thousand hours of testing, two types of pivotless breaker were developed.

The pivotless type of breaker has shown its superiority over the lever type from the very first experimental model. Three conventional lever types of magneto were bench-tested at 3,800 r.p.m. and ran continuously without adjustment for 15, 100 and 116 hours respectively. In each case, the contacts were worn sufficiently to require either refacing or renewal. The first pivotless type ran for more than 500 hours on the same set of contacts at 4,200 r.p.m. and was still in good condition, the total wear on the points being about 0.008 inch. A pair of pivotless breakers on a vertical double magneto recently ran for 1,015 hours at 4,200 r p.m. without trouble.

Delco Battery Ignition System.—The Delco battery ignition system was first applied to the Liberty engine because no magnetos were in production that would give the proper current delivery on account of the sparks not being spaced at exact intervals, due to the small angle between the cylinders, and the battery distributor could be easily adapted to this irregular spark distribution without loss of spark intensity. It was found that certain advantages were realized by the use of the Delco distributor which had the following features:—

(1) Direct drive from the camshaft
(2) Ignition coil integral with the distributor head
(3) Double breaker arms
(4) Auxiliary contact arm to prevent sparking on reverse rotation
(5) Carbon brush type distributor
(6) Spark advance effected by shifting the entire distributor.

The use of a carbon brush in the distributor necessitates frequent cleaning of the distributor track to remove the oil and carbon. It has also been found that moisture in the coil insulation that had not been entirely removed during the process of manufacture greatly reduces the energy of the spark, when the coil reaches a temperature of more than 150 degrees Fahrenheit, sometimes causing scattered missing and roughness of the engine. This has been a very obscure trouble and, as usual, the condenser has been blamed for much of it. The usual method of cure has been to put on a new distributor. More or less trouble has always been caused by oil leakage, particularly on inverted engines. To overcome this trouble and to improve the reliability of the distributor, the Materiel Division, with the co-operation of the Delco Co., has worked out the following modifications, for converting the carbon brush type of distributor into what is called the Liberty Delco air-gap type of distributor:

(1) Carbon brush replaced with metal pin to give an air-gap distributor
(2) All moisture removed from coils by careful vacuum drying at 250 degrees Fahrenheit, and revarnishing the exterior
(3) Upper rotor-shaft bearing replaced with new bearing having integral oil-retaining washer
(4) Bakelite coil-housing and aluminum distributor-cup replaced with new parts accurately machined to assure concentric distributor track
(5) Suitable oil-drain holes provided to remove the excess oil
(6) Fireproof vent-holes in distributor cup.

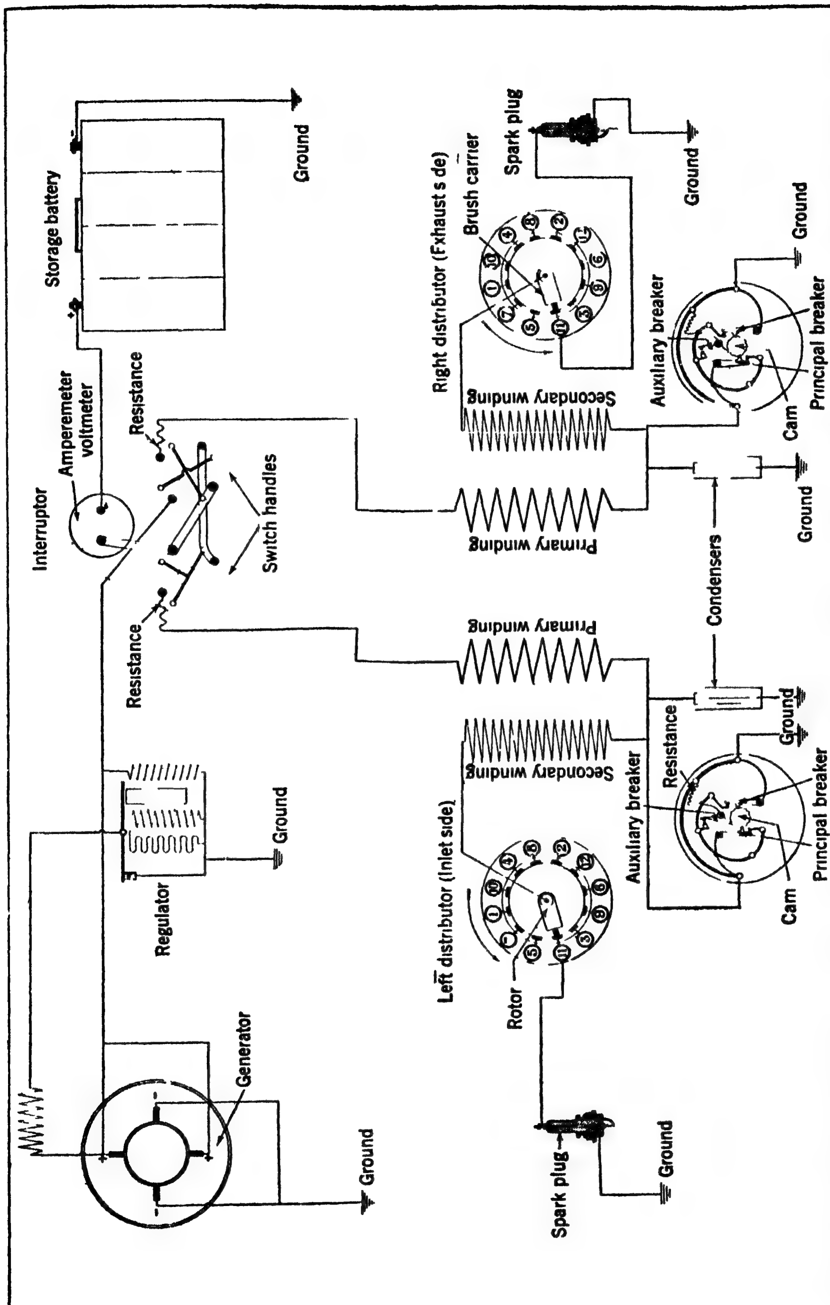


Fig. 192.—Diagrams Showing the Wiring Connections of the Delco Ignition System Used on Lorraine Twelve-Cylinder "Vee" Engine.

These changes have virtually eliminated troubles from carbon on the track and oil leakage. Four distributors were run on the bench for 350 hours without attention and no "tracking" was experienced. Service tests in flight have shown a similar freedom from carbon troubles.

Delco Wiring Diagrams.—The wiring diagram at Fig. 192, which is simplified by only showing one secondary cable and plug connected to each distributor head clearly shows the electrical principle involved in the Delco system. Current from an engine driven generator, mounted as shown at Fig. 193 goes through a cut-out relay in one system or through a voltage regulator in another and is used to charge a storage battery of the twelve-

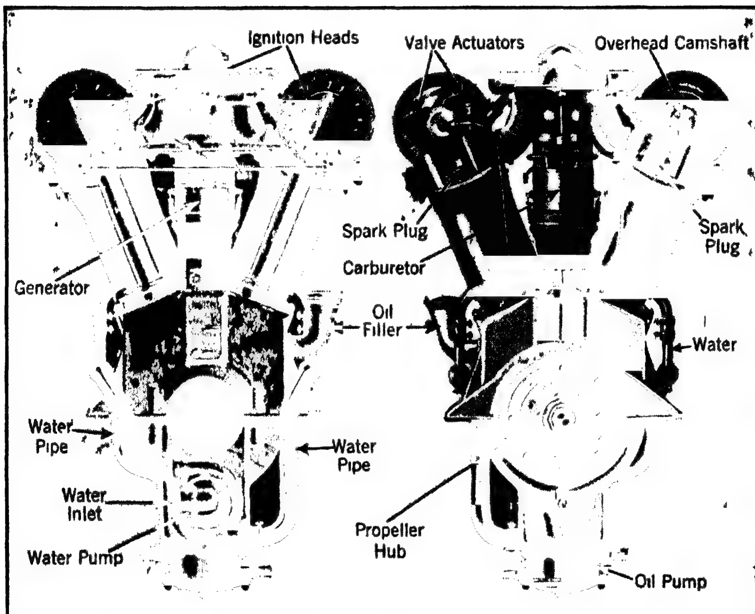


Fig. 193.—Rear and Front Views of Liberty Aviation Engine, an American Development that was Produced in Large Quantities in 1918. Note Generator Mounting as Shown at Left of Illustration.

volt type. The combined distributor and contact maker is mounted on and driven by an extension of the camshaft in some systems and by separate gear drive from any convenient point in other engines. Induction coils intensify the primary current, the high-tension current being supplied to the distributor brush in the usual manner. From that member, it is led to the various sparkplugs in their proper firing order. The distributor used on Packard 2A1500 and 2A2500 is shown at Fig. 194 and the wiring diagram of the Packard ignition system is shown at Fig. 195. A combined switch and amperemeter is used, there being two switch handles, one for intake plugs, the other to control the exhaust side sparkplugs. Either set or both sets together may be employed.

Characteristics of Ideal Ignition System.—Based on the requirements of the types of military and commercial aircraft now in service and proposed for the future, Mr. Shoemaker believes that the ideal ignition system should

have the following characteristics:

(1) A rugged light compact source of sparks
(2) Flange mounting
(3) Bearings large enough to allow a direct splined shaft or gear drive
(4) Complete enclosure of the ignition system in a metallic housing conforming to the space available on the engine
(5) Secondary cables carried in substantial metallic housings preferably built into the cylinder-block and connected directly to the distributor housing, without flexible braid or tubing, thus providing radio shielding and mechanical protection and eliminating the fire hazard
(6) Ignition drive from the propeller end of the crankshaft to eliminate drive stresses
(7) Ignition for starting direct from the "running" magneto without the use of booster magneto
(8) Electrical insulation for supercharging to at least 20,000 feet.
(9) A normal life, without lubrication, adjustment or cleaning, greater than the time between the major overhauls of the engine
(10) Accessibility in the airplane for inspection of breaker and distributor
(11) Easily removable and interchangeable breakers, condensers, coils, and distributor parts
(12) Standardization of the basic parts of a given make, such as main frames, coils, condensers, rotors, bearings, breakers, and the like, to make possible the use of the same parts with all types of engines.

Many of these ideal features are yet to be realized, but the following may be considered as having reached a usable state of development:

(1) A reduction in the weight of the ignition system of from 25 to 40 per cent
(2) An increase in the satisfactory operating speed of magnetos of at least 100 per cent
(3) Extension of the life of the magneto breaker at least five times
(4) Elimination of the fire hazard
(5) Direct drive without flexible coupling
(6) Flange mounting
(7) Air-gap distributor.

In conclusion it must be emphasized that the development of ignition equipment specially adapted to aircraft engines has just begun. So far, the conventional single magneto has the same general appearance as the first magnetos built. A glance at almost any aircraft engine gives a distinct impression that the ignition equipment has been hung on after the engine was finished. The double magneto is an attempt to adapt the ignition to the aircraft engine, but there are still possibilities of smaller and more compact units than any now available and of more serviceable installation on the engine.

Radio Shielding.—In practice, it is necessary to surround the entire ignition system with well-bonded metallic housings that enclose the magneto, distributors, ignition cables, and as much of the sparkplugs as possible to prevent interference with the airplane radio system. The ignition-cable

tubes must be grounded at each cylinder, as high-frequency static charges will build up on the tubes and arc across to neighboring conductors if the tubes are not grounded at least every twelve inches. On many engines, the use of radio shielding will necessitate a re-arrangement of the accessories to allow the installing of the necessary manifolds and the proper housing of the distributor. The question naturally arises, why not use the cowling on all-metal ships as a means of shielding. This has been tried, but the necessity for bonding wires between the various sections of the cowling makes the inspection and care of the powerplant very tedious, and the shielding is only about one-half as effective as is the manifold type.

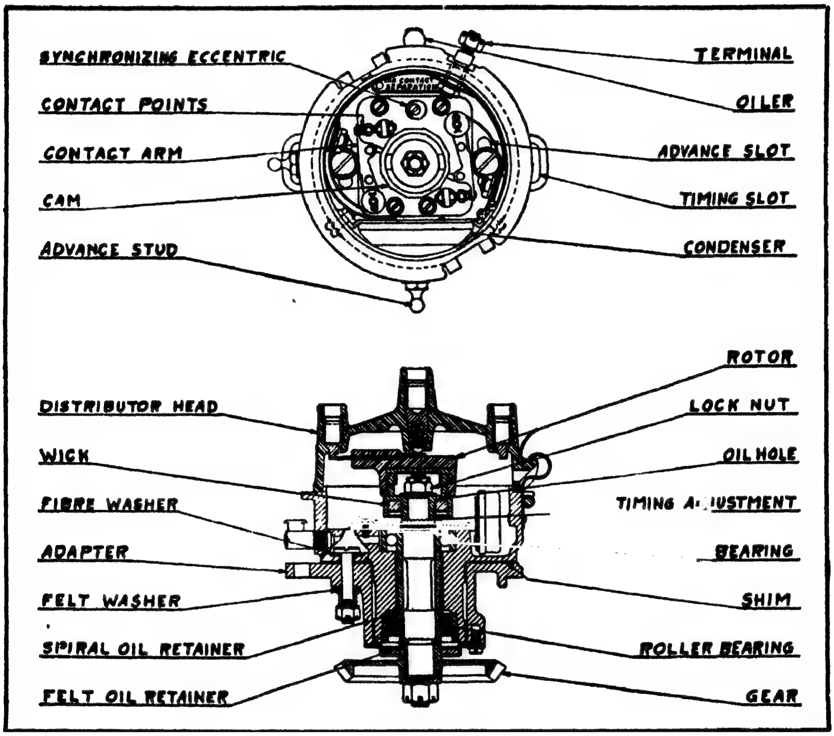


Fig. 194.—Delco Battery Distributor Assembly Used on Packard Model 2A-15000, and 2A-2500 Aircraft Engines.

The effect of shielding on the ignition is of considerable importance also. Mr. Shoemaker states that the addition of grounded surfaces close to the secondary cables throughout their length increases their electrical capacity from 50 to 100 per cent. In some types of ignition in which the energy from the coil is limited, this increase in secondary capacity may be sufficient to prevent the coil from charging this capacity up to the required sparking-voltage, and missing will occur. When the spark energy is ample, however, the additional capacity causes the spark to be "snappier," by increasing the initial capacity component of the spark. In one of the first shielded installations on the Liberty-12 engine, the secondary cables were each covered with a tightly woven copper-braid, and the entire bundle of wires was

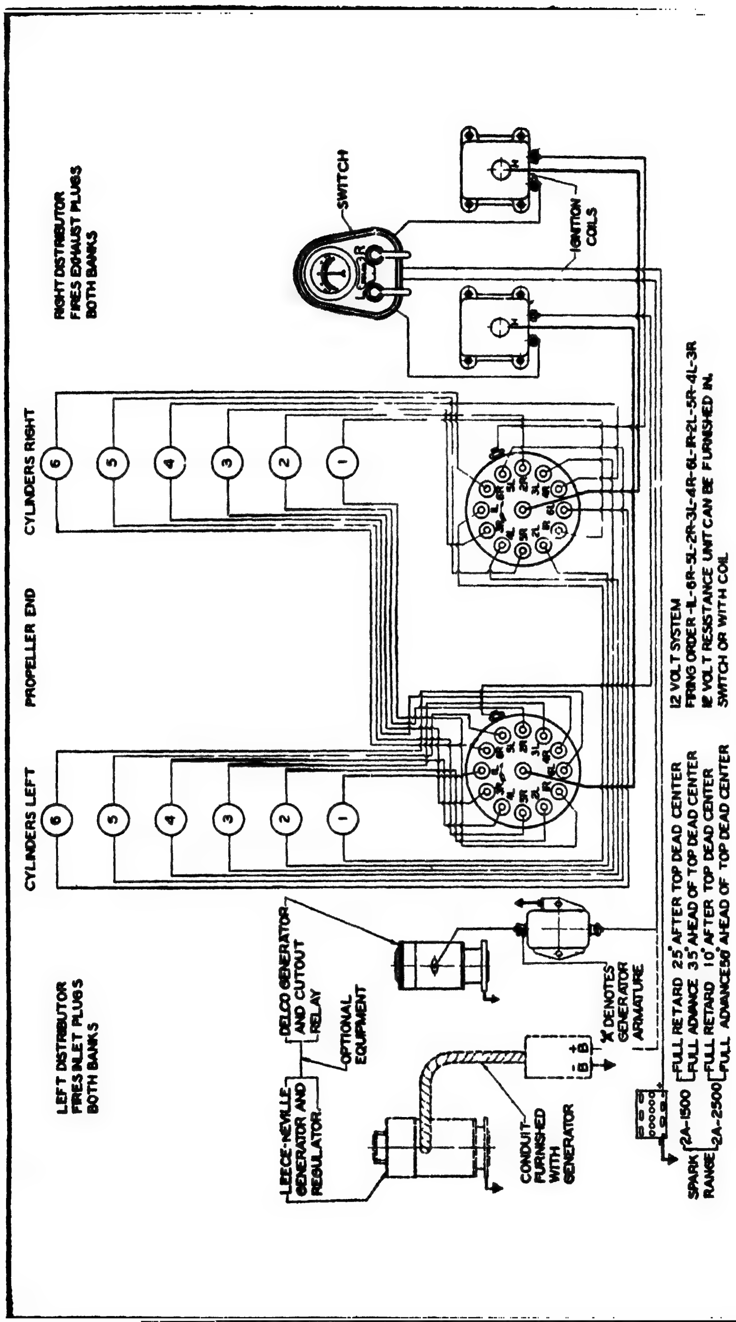


Fig. 195.—Wiring Diagram of Packard Aircraft Engines Using Delco Battery Ignition System.

grounded to the engine at frequent intervals. This increased the secondary capacity to such an extent that missing occurred in service. By removing the copper braid from the cables and using metal covers over the distributors that were connected to the manifolds in the Vee of the engine, the capacity was greatly reduced and no further trouble was encountered. The following interesting examples illustrate some of the obscure sources of radio interference.

An inspection of several airplane engines run on the ground at night showed the presence of many coronas and static discharges from various metal parts around the secondary cables. These were particularly pronounced with magneto ignition. Under very favorable conditions, these static discharges will ignite gasoline. Several cases were found in which

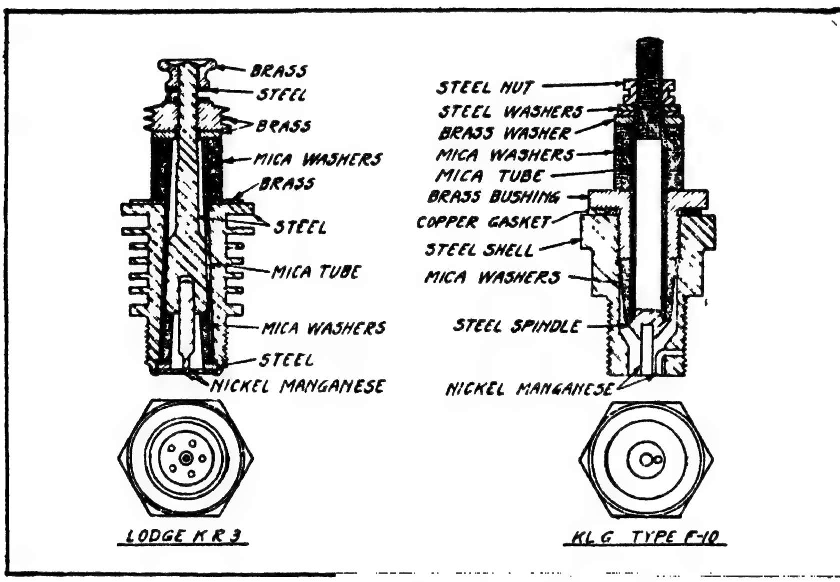


Fig. 196.—Mica Insulated Sparkplugs for Aviation Engine Ignition.

the ignition sparks themselves jumped to ground where the clearance was too small. The obvious remedy for such troubles is to enclose the secondary distributing system completely with well-grounded metal manifolds and to provide sufficient insulation, or clearance, around the sparkplug terminals. This also meets the requirements of radio shielding, as previously outlined. In general, the problems of eliminating fire hazards due to the ignition system are mostly mechanical and have little or no effect on the electrical design of the ignition system.

Sparkplug Design and Application.—With the high-tension system of ignition the spark is produced by a current of high voltage jumping between two points forming a gap which breaks the complete circuit, which would exist otherwise in the secondary coil and its external connections. The sparkplug is a simple device which consists of two terminal electrodes carried in a suitable shell member, which is screwed into the cylinder.

Typical sparkplugs are shown in section at Figs. 196 to 199 inclusive and the construction can be easily understood. The secondary wire from the coil is attached to a terminal at the top of a central electrode member, which is supported in a bushing of some form of insulating material. The AC type shown at Fig. 198 employs a moulded porcelain as an insulator, while the Oleo depicted at Fig. 199 uses a bushing of mica. The insulating bushing and electrode are housed in a steel or brass body, which is provided with a screw thread at the bottom, by which means it is screwed into the combustion-chamber.

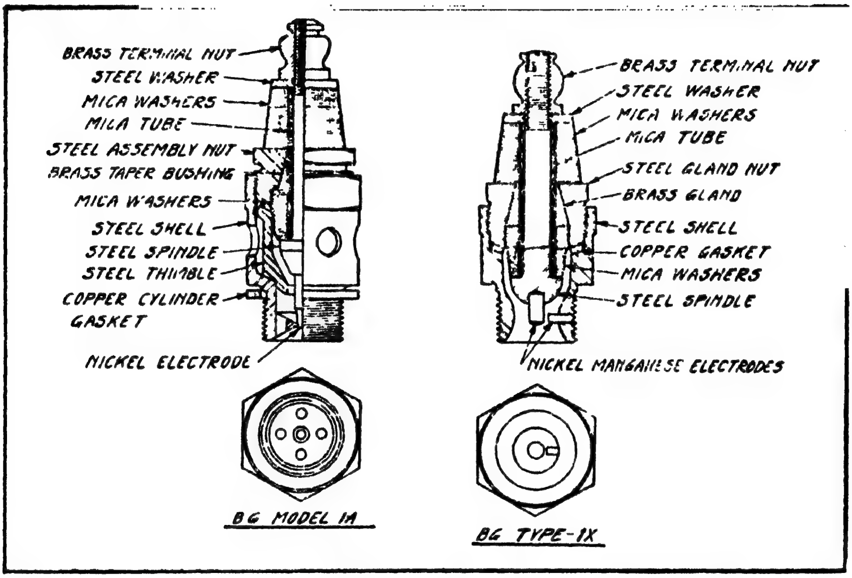


Fig. 197.—Practical Sparkplugs Specially Designed for Aviation Engines.

When porcelain is used as an insulating material it is kept from direct contact with the metal portion by some form of yielding packing, usually asbestos. This is necessary because the steel and porcelain have different coefficients of expansion and some flexibility must be provided at the joints to permit the materials to expand differently when heated. The metal body of the plug which is screwed into the cylinder is in metallic contact with it and carries one or more sparking points which form one of the terminals of the air gap over which the spark occurs. The current entering at the top of the plug cannot reach the ground, which is represented by the metal portion of the engine, until it has traversed the full length of the central electrode and overcome the resistance of the gap between it and the terminal point on the shell. The porcelain bushing is firmly seated against the asbestos packing by means of a spun over piece at the top of the plug body which sets against a flange formed on the porcelain.

The mica plug of Oleo design shown at Fig. 199 is somewhat simpler in construction than that shown at Fig. 197 and designated as the B. G. Type IX. The mica core which keeps the central electrode separated from the steel body is composed of several layers of pure sheet mica wound

around the steel rod longitudinally, and hundreds of stamped mica washers which are forced over this member and compacted under high pressure with some form of a binding material between them. Porcelain insulators are usually moulded from high-grade clay and are approximately of the shapes desired by the designers of the plug. The central electrode may be held in place by mechanical means such as nuts, packings, and a shoulder on the rod, as shown at Fig. 196, which outlines the construction of the Lodge KR3 sparkplug. Another method sometimes used is to cement the electrode in place by means of some form of fire-clay cement. Whatever method of fastening is used, it is imperative that the joints be absolutely tight so that no gas can escape at the time of explosion. Porcelain is the material most widely used if mica is not employed because it can be glazed

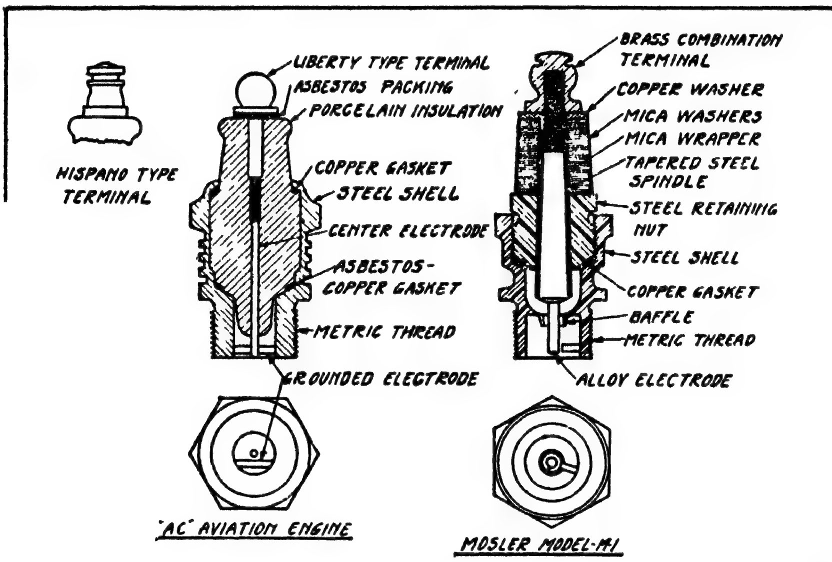


Fig. 198.—The AC Sparkplug at Left Uses Porcelain Insulation, while the Mosler Shown at Right has Mica Insulation.

so that it will not absorb oil, and it is subjected to such high temperature in baking that it is not liable to crack when heated. The sparkplugs may be screwed into any convenient part of the combustion-chamber, the general practice being to install them in the head over or adjacent to the inlet valves, or in the side of the combustion-chamber, so the points will be directly in the path of the entering fresh gases from the carburetor. Of course, when double ignition is used, one set of plugs must be placed near the exhaust valves.

Other insulating materials sometimes used are glass, steatite (which is a form of soapstone) and lava. Mica and porcelain are the two common materials used because they give the best results. Glass is liable to crack, while lava or the soapstone insulating bushings absorb oil. The spark gap of the average plug is about 1/32 of an inch for coil ignition and from .015 inch to .030 inch (fifteen to thirty one-thousandths of an inch) when used

in magneto circuits. A simple gauge for determining the gap setting is the thickness of an ordinary visiting card for magneto plugs, or a space equal to the thickness of a worn dime for a coil plug. This is but a rough approximation and the best practice is to use proper thickness gauges and set gaps as recommended by engine manufacturers. The insulating bushings are made in a number of different ways, and while details of construction vary, sparkplugs do not differ essentially in basic design. The dimensions of the standardized plug recommended by the S. A. E are shown at Fig. 200.

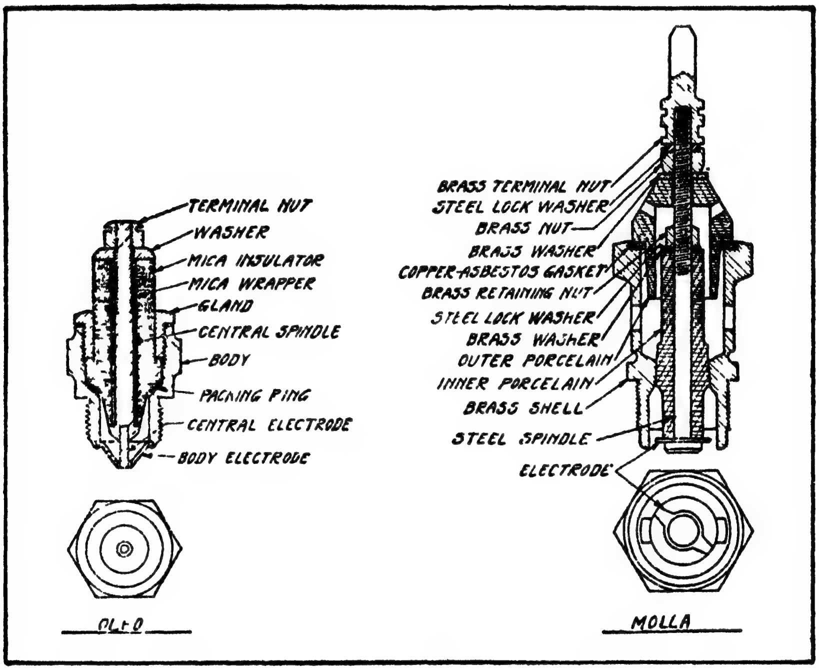


Fig. 199.—The Oleo Plug at Left Uses Mica Insulation. The Molla Plug at the Right is a Double Insulator Type with Passages for Air Circulation.

It is often desirable to have a water-tight joint between the high-tension cable and the terminal screw on top of the insulating bushing of the sparkplug, especially in applications where plugs are exposed to the weather, as on radial cylinder air-cooled engines. Plugs have been made that are provided with an insulating member or hood of porcelain, which is secured by a clip in such a manner that it makes a water-tight connection. Should the porcelain of a conventional form of plug become covered with water or dirty oil, the high-tension current is apt to run down this conducting material on the porcelain and reach the ground without having to complete its circuit by jumping the air gap and producing a spark. It will be evident that wherever a plug is exposed to the elements, which is often the case in airplane service, that it should be protected by an insulating hood which will keep the insulator dry and prevent short circuiting of the spark. The

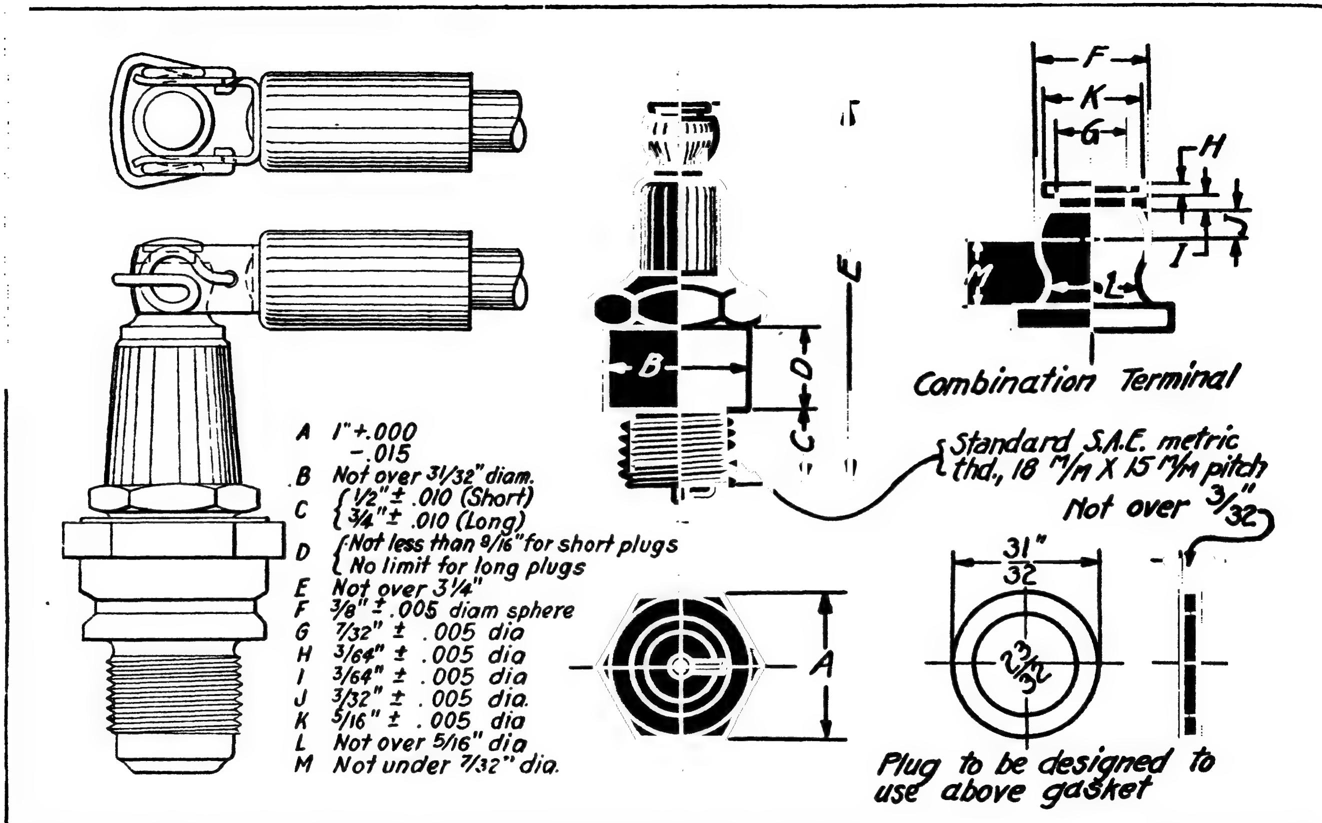


Fig. 200.—Dimensions of Standard S.A.E. Sparkplug for Aviation Engine Ignition. The Safety Lock Sparkplug Terminal Shown at the Left Will Not Come Off Until Released Manually. No Tools are Required for its Operation.

same end can sometimes be attained by slipping an ordinary rubber nipple over the porcelain insulator of any conventional plug and bringing up one end over the cable, taping it in place.

Attention is directed to the B. G. Model 1A at Fig. 197 and the Molla at Fig. 199. In the former, there are holes in the shell that permit cooling air to circulate through the plug body and around the insulator. The Molla plug provides for a more positive cooling of the insulator by having both inlet holes in the shell and outlet holes in the top porcelain member.

Two-Spark Ignition.—On most aviation engines, especially those having large cylinders, it is sometimes difficult to secure complete combustion by using a single sparkplug. If the combustion is not rapid the efficiency of the engine will be reduced proportionately. The compressed charge in the cylinder does not ignite all at once or instantaneously, as many assume, but it is the strata of gas nearest the plug which is ignited first. This in turn sets fire to consecutive layers of the charge until the entire mass is aflame. One may compare the combustion of gas in the gas-engine cylinder to the phenomenon which obtains when a heavy object is thrown into a pool of still water. First a small circle is seen at the point where the object has passed into the water, this circle in turn inducing other and larger circles until the whole surface of the pool has been agitated from the one central point. The method of igniting the gas is very similar, as the spark ignites the circle of gas immediately adjacent to the sparking point, and this circle in turn ignites a little larger one concentric with it. The second circle of flame sets fire to more of the gas, and finally the entire contents of the combustion-chamber are burning.

While ordinarily combustion is sufficiently rapid with a single plug so that the proper explosion is obtained at moderate engine speeds, if the engine is working fast and the cylinders are of large capacity more power may be obtained by setting fire to the mixture at two different points instead of but one. This may be accomplished by using two sparking-plugs in the cylinder instead of one, and experiments have shown that it is possible to gain from 25 to 30 per cent in motor power at high speed with two sparkplugs, because the combustion of gas is accelerated by igniting the gas simultaneously in two places. The double-plug system on airplane engines is also a safeguard, as in event of failure of one plug in the cylinder the other would continue to fire the gas, and the engine will continue to function regularly though not at full power delivery.

In using magneto ignition some precautions are necessary relating to wiring and also the character of the sparkplugs employed. The conductor should be of good quality, have ample insulation, and be well protected by metal housings from accumulations of oil, which would tend to decompose rubber insulation. It is customary to protect the wiring by running it through the conduits of fiber or metal tubing lined with insulating material. Multiple strand cables should be used for both primary and secondary wiring, and the insulation should be of rubber at least $\frac{3}{16}$ inch thick.

The sparkplugs formerly used for battery and coil ignition in automobiles cannot always be employed in an airplane engine when a magneto is fitted. The current produced by the mechanical generator has a greater amperage and more heat value than that obtained from transformer coils

excited by battery current. The greater heat may burn or fuse slender points used on some battery plugs and heavier electrodes are needed to resist the heating effect of the more intense arc. While the current has greater amperage it is not of as high potential or voltage as that commonly produced by the secondary winding of an induction coil, and it cannot overcome as much of a gap. Manufacturers of magneto plugs usually set the spark points from .015 inch to .025 inch apart. An efficient magneto plug has a plurality of points so that when the distance between one set becomes too great the spark will take place between one of the other pairs of electrodes which are not separated by so great an air space. This effect is sometimes obtained by using a disc surrounding the central electrode, the spark jumping anywhere in the annular gap thus formed.

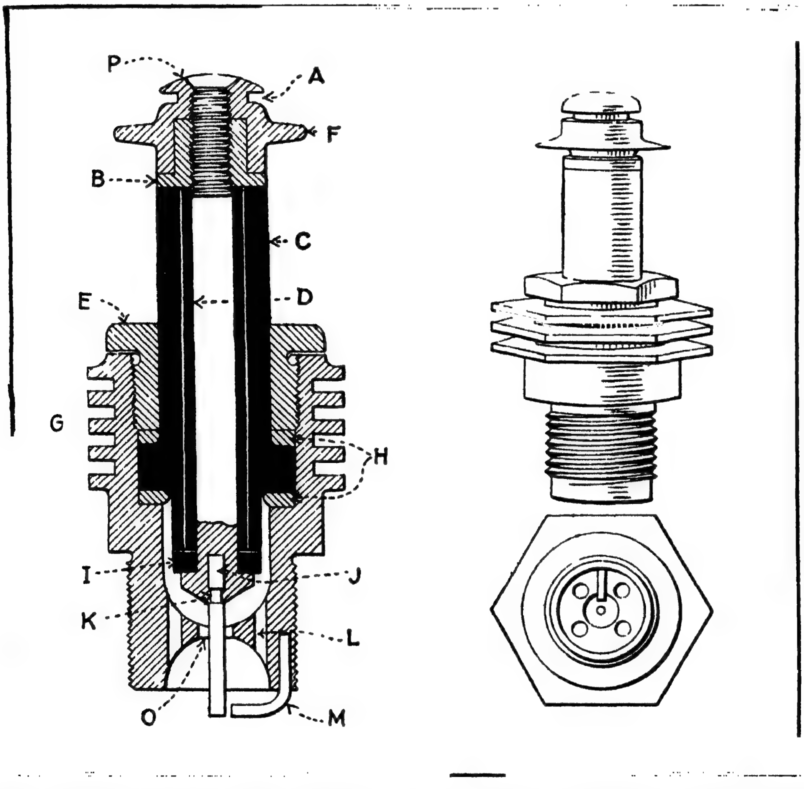


Fig. 201.—Special Mica Plugs Designed for Aviation Engines Have Air-Cooling Flanges on Body.

Airplane work calls for special construction of sparkplugs, owing to the high compression used in the engines and the fact that they are operated on open throttle practically all the time, thus causing a great deal of heat to be developed. The plug shown at Fig. 201 has been devised especially for airplane engines and automobile racing powerplants. The core C is built up of mica washers, and has square shoulders. As mica washers of different sizes may be used, and accurate machining, such as is necessary

with conical clamping surfaces, is not required, the plug can be produced economically. The square shoulders of the core afford two gasket seats, and when the core is clamped in the shell by means of check nut E, it is accurately centered and a tight joint is formed. This construction also makes a shorter plug than where conical fits are used, thus improving the heat radiation through the stem. The lower end of the shell is provided with a baffle plate O, which tends to keep the oil away from the mica. There are perforations L in this baffle plate to prevent burnt gases being pocketed behind the baffle plate and preigniting the new charge. This construction also brings the firing point out into the firing chamber of the engine, and has all the other advantages of a closed-end plug. The stem P is made of brass or copper, on account of their superior heat conductivity, and the electrode J is swedged into the bottom of the stem, as shown at K, in a secure manner. The shell is finned, as shown at G, to provide greater heat radiating surface. There is also a fin F at the top of the stem, to increase the radiation of heat from the stem and electrode. The top of this finned portion is slightly countersunk, and the stem is riveted into same, thereby reducing the possibility of leakage past the threads on the stem. This finned portion is necked at A to take a slip terminal.

In building up the core a small section of washers, I, is built up before the mica insulating tube D is placed on. This construction gives a better support to section I. Baffle plate O is bored out to allow the electrode J to pass through, and the clearance between baffle plate and electrode is made larger than the width of the gap between the firing points, so that there is no danger of the spark jumping from the electrode to the baffle plate.

This plug will be furnished either with or without the finned portion, to meet individual requirements. The manufacturers lay special stress upon the simplicity of construction and upon the method of clamping, which is claimed to make the plug absolutely gas-tight.

All of the plugs illustrated have given good service. The AC aviation engine porcelain insulation plug has been widely used. The Oleo plug gives very good results in air-cooled cylinders, and the BG Type IX, Mosler Model MI and Lodge KR3 have also given good service in both air-cooled and water-cooled engine installations.

Electric Generators for Airplanes.—Exacting requirements for the generation of electric current for motorcoaches, airplanes and dirigibles, and how these needs are met by the manufacturers of generators, are described by B. M. Leece, M.S.A.E., in a paper read before the Chicago Section of the Society and reported in the June, 1928, issue of the *S. A. E. Journal.* Concurrently with the development of automotive vehicles, the installation of electrical equipment has grown and greater and greater demands have been made upon generators. Requirements change rapidly with development in the powerplant and in the general design of vehicles, so that manufacturers of electric generators are constantly faced with new conditions. A high degree of reliability in constant operation is also required of the generator on the latest airplanes and airships, for ignition, engine starting, radio communication, and lighting. Electric generators are expected today to operate under widely varying specifications as to capacity, voltage, speed, load, and

temperature. Under present commercial conditions, the generator capacity is often forced to the limit of material characteristics, and operation is continuous; therefore, the rating of such generators is a matter of concern to both the manufacturer and the user. Whereas the third-brush, or current-controlled generator gave fairly satisfactory service on private automobiles, charging the battery at an approximately constant rate in amperes, it was found undesirable when the equipment is to be used continuously, because the excess energy heats the electrolyte and plates after the battery is fully charged. In commercial applications it is desirable to have the generator start charging the battery at a low driving speed and to be able to continue under full load up to the maximum speed of the vehicle. Hence the voltage-regulated type was developed, which supplies energy at an approximately uniform voltage and at a charge rate that tapers automatically in amperage. The characteristics of this type are discussed and compared with those that are inherent in the third-brush type.

The airplane is being developed rapidly and requires a high degree of reliability in constant operation. Radio signalling on railroad trains presents another problem. Large dirigibles, such as are being built, require enormous amounts of electrical energy to be generated from a multiplicity of powerplants and transmitted for distances of several hundred feet. Recently, one company has been working on the design of generating equipment for a dirigible approximately 700 feet long. The design calls for eight generators, running in parallel, to meet the voltage requirements of the storage battery. On aircraft equipped for night flying, powerful landing-lights are necessary. These draw a very large amount of current and are one of the great factors influencing the size of the generator and battery which must be used. Two of these lamps are used on each airplane, each drawing approximately 45 amperes when in operation. On B1-type airplanes the lamps are located on the upper wings on some installations and on the lower wings on others. In the new Curtiss Condor airplane, the two landing-lights are centrally mounted. Two generators operating in parallel are used, one on each of the two engines. This is one of the few installations of its kind in existence.

Third-Brush Regulation Not Well Adapted to Airplane Work.—The third-brush or current-controlled generator has been used on passenger cars for a number of years. There have been and still are some limitations to its performance but, in general, it has been found to be a fairly satisfactory generator for automobile service, all things considered. When adjusted for rated output at required speed, the third-brush generator will charge a battery at an approximately constant rate in amperes no matter what may be the state of charge of the battery. In fact, the third-brush machines give slightly higher outputs when the battery is up. This constant rate of charge has been rather undesirable when the equipment is to be used continuously, for a prolonged high charging rate will charge the battery fully and the excess energy which the generator furnishes can be absorbed only in heating the electrolyte and plates of the battery, causing the water to be driven off. There is danger that the batteries may become dry when a third-brush generator is used continuously for extended periods unless the third brush is adjusted normally for a low rate. If this is done, the chances

are that the battery will not be sufficiently charged. Should a battery terminal become loosened and the generator be operated without the battery connected, its terminal voltage is uncontrolled and is proportional to speed, and this often results in a rise in voltage which burns out the lamps connected or blows the fuse usually provided in the field circuit. With the field circuit open, the generator is protected against the burning out of the armature winding but it is also inoperative. If battery terminals become badly corroded, thereby introducing added resistance into the external circuit of a third-brush machine, the generator automatically delivers more energy to overcome the increased resistance. This may result in current high enough to endanger the windings of the machine or blow out the lamps or line fuses.

Strictly speaking, however, the voltage-regulated generator, as constructed at present by several companies, is a modified form of the constant-potential generator. But the result is that the voltage-regulated generator produces a charging rate in amperes which is automatically tapering. If the battery is in a discharged condition, the charging rate will be comparatively high. If the battery is fully charged, the charging rate will be low. As a discharged battery is charged, its voltage slowly rises. The difference between battery voltage and charging voltage decreases, which results in a decreasing or tapering charging rate in amperes. Speed and load characteristics of the voltage-regulated type of generator are more suitable for commercial application. Generators can be built which will carry full load at normal speeds and continue to furnish almost their full rated energy up to the limit of speed of the engine. In comparing the operation of the third-brush type of generator with that of the voltage-regulated generator, it must be remembered that the first charges a battery at a fairly uniform rate and that the second charges in proportion to the state of charge of the battery.

It should be noted that there is a considerable difference between the two charging rates as the charge proceeds; that is, the third-brush type of generator furnishes to the battery energy which is at times in excess of its needs. It may charge more rapidly, but this is not an unalloyed virtue because the electrolyte should not be overheated. If the voltage-regulated generator charges a battery with less input of energy, an appreciable saving is effected which can be translated into horsepower and money.

Voltage-Regulated Generators.—Voltage-regulated generators are divided roughly into the following three main classes: (*a*) those of low or medium capacity in which an extremely low balancing speed is not required, (*b*) those of higher capacity in which a low balancing speed is desirable, and (*c*) high-capacity generators for high-speed service.

In classes (*a*) and (*c*) it is customary to retain the third brush as a means of limiting the current output of the generator. A voltage-control unit is applied, which causes the generator to function as a voltage-regulated machine with the inherent overload protection provided by the third brush.

Class (*b*) generators are straight shunt-wound machines having no third brush. These can be built to balance the battery at lower speeds; but they require some means of overload protection, as they will furnish more cur-

rent than the armature winding can be made to conduct without overheating. It is usual to protect these generators by some external means. In some instances a fuse has been used; in others, an automatic overload circuit-breaker is employed which trips open the load circuit on overloads and has to be reset by hand. A third alternative, and the one most generally used, is to incorporate a current-limiting element in the control unit to limit automatically the current which the generator can deliver and

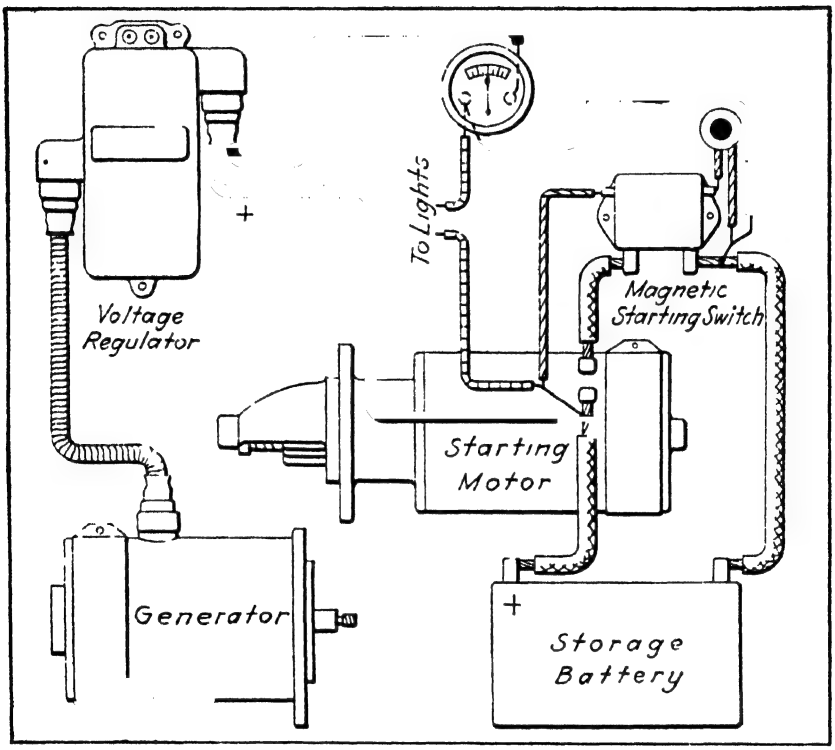


Fig. 202.—Wiring Diagram Showing Method of Connecting Voltage Control Unit at Generating Circuit.

which does not have to be reset by hand. Such current-limiting devices have been used on each of the three classes of generator.

Voltage regulation has been found to be essential to the satisfactory operation of generators for aircraft application. In this service the engine runs at a fairly constant high speed and, especially in cross-country flying, for long periods. In addition, it is highly desirable to have a source of approximately constant-potential electrical energy for the operation of radio and other devices. Technicians of the Army Air Service were among the first to appreciate the advantages of voltage regulation as applied to their problems, and since the war a continuous and uninterrupted program of development has been carried on which has resulted in a line of generators and control units for aircraft. When slightly modified, these serve admir-

ably on commercial airplanes. The Air Corps requirements, which include radio operation, are much more rigid than those necessary for commercial aviation.

AIRCRAFT GENERATORS AVAILABLE

Type	Generator Speed, R.P.M. Minimum to Carry Rated Load	Generator Speed, R.P.M. Normal	Rated Potential, Volts	Rated Line-Current, Amp.	Maximum Weight, Generator Only, Lb.	Outside Diameter, In.	Direction of Rotation at Drive End
B-1	2,000	2,250	15	25	28	5.0	Clockwise
C-1	2,000	2,250	15	50	44	6.0	Clockwise
D-1	2,250	2,750	15	25	20	4.5	Counter-clockwise
E-3	2,250	2,750	15	50	35	5.5	Counter-clockwise
E-4	2,250	2,750	15	50	35	5.5	Clockwise
F-1	2,000	2,250	15	15	24	4.5	Clockwise
G-1	2,250	2,750	15	15	16	4.0	Counter-clockwise
G-2	2,250	2,750	15	15	16	4.0	Clockwise

Aircraft material has been emphasized in this discussion for several reasons. First, it is interesting to everyone today. In electrical application, air transport is leading other branches of automotive engineering. Aircraft equipment is a good proving laboratory, for the tests are rigorous and the requirements exacting. Aviation applications are proving the correctness of the principle of voltage regulation. If electrical loads are to increase further and the engine builders continue to restrict the space available for the generator, the parallel operation of generators may assume more importance in the future. Numerous instances have arisen in which a motorcoach, for example, required energy in excess of that which the generator could furnish, because a large enough generator could not be mounted in the space available.

On a large dirigible that is being designed in this country it is proposed to carry eight powerplants in separate compartments. As the airship is more than 700 feet long, the problem of transmitting electrical energy without serious loss must be considered. It is proposed that eight 110-volt generators be operated in parallel on this installation to supply one battery, all cells of which will be centrally located. Generators used in aircraft applications may be driven by direct mechanical connection with the engines or may be operated by air-driven propellers. In some large airliners, a generator set consisting of a small air-cooled engine and dynamo may be used to make the radio independent of the main powerplants or of the motion of the machine through the air so it can be used in event of a forced landing.

Voltage regulators manufactured today are reliable to a high degree and function automatically. In general, it is a good rule to make sure that they are properly adjusted to meet requirements and then to let them alone.

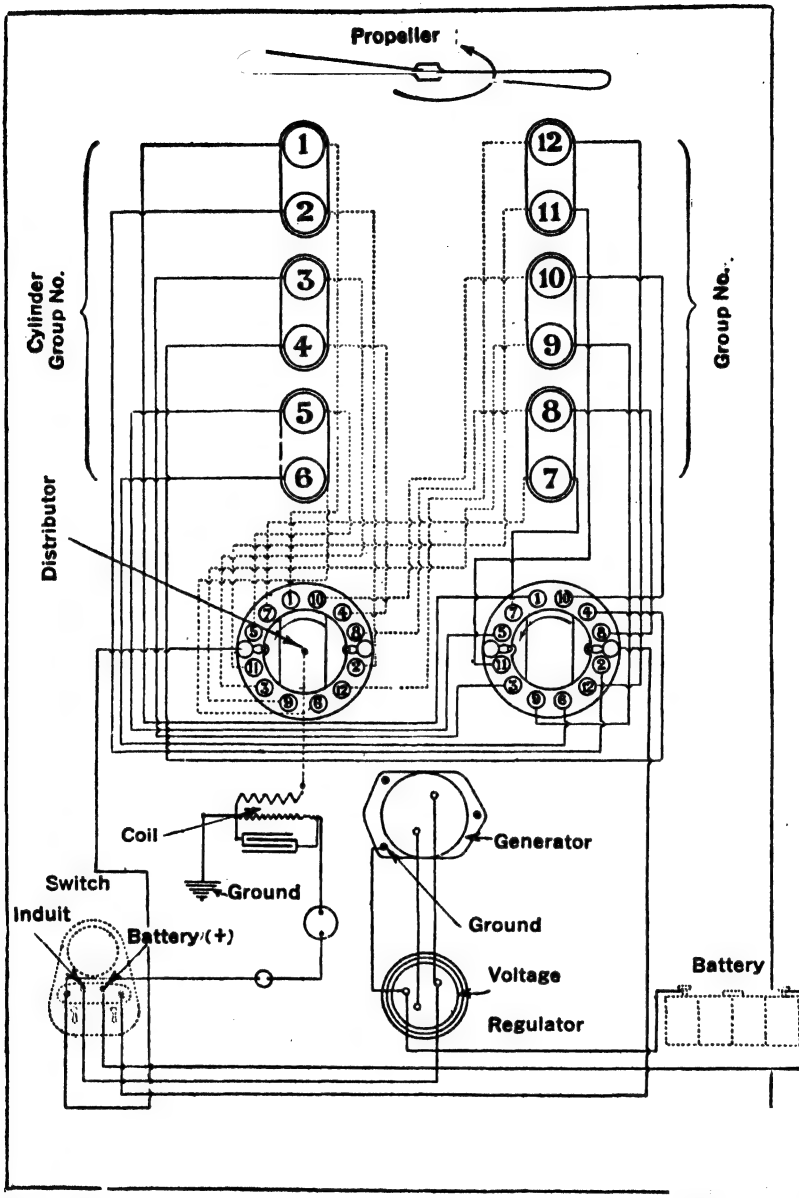
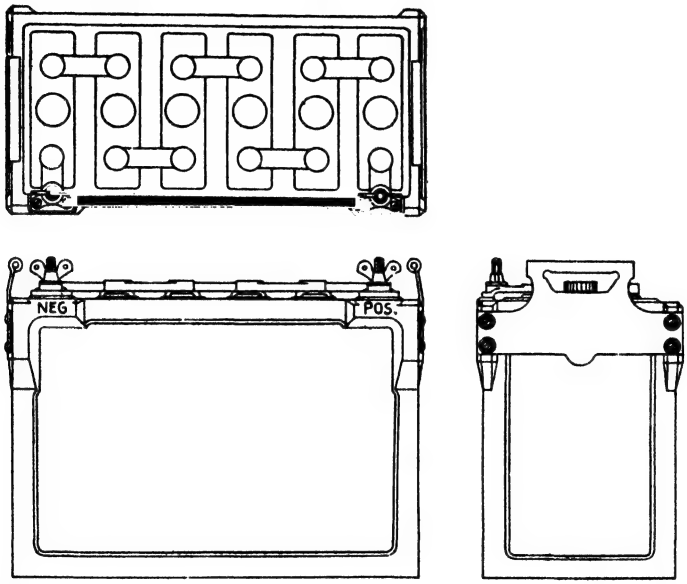


Fig. 203.—Wiring Diagram Showing Delco System of Twelve-Cylinder Lorraine Aviation Engine.

In a voltage-regulated system, if the battery is fully charged the charging rate in amperes will be low. Many have assumed that a low ammeter reading meant that the generator was not producing enough current and have raised the voltage and the charging rate, which naturally resulted in an excessively high rate with all of its attendant troubles. Instances have oc-

curred in which the voltage has been raised, through ignorance, to more than twenty volts on a twelve-volt system, despite every effort that could be made to educate the user to the characteristics of voltage-regulated generating equipment. The way the voltage regulator is wired into the circuit of an electric starting and lighting system is outlined at Fig. 202. The use of a voltage regulator in the Delco ignition system of the Lorraine engines is outlined in the very complete wiring diagram presented at Fig. 203.

Experience has also shown that it is better to adjust the charging so that the battery is cycled frequently. This procedure has the sanction of the battery manufacturers, as longer battery life results. It is better to allow a battery to discharge somewhat and then to recharge, than it is to keep it in a fully charged condition at all times. It has also been found



Battery No.	Number of Cells	Minimum Capacity at 5-Hr. Rate, Amp.-Hr.	Minimum Current for 20 Min., Amp.	Maximum Over-All Dimensions, In.			Maximum Weight Wet, Lb.
				Length[1]	Width	Height	
32	6[a]	29½	50	8$\frac{1}{16}$	7¼	11¼	38
34	6[a]	43	75	11½	7¼	11¼	53
36	6[a]	65	113	14½	7¼	11¼	70

[1] The over-all end-to-end length includes handles, not hold-down devices. The handles and the hold-down devices shall be attached only to the ends of the case.
[a] Side-to-side assembly of cells.

Fig. 204.—Standard Dimensions for Storage Batteries for Airplane Use.

that in many cases it is desirable to charge the battery at higher rates in the cold months than in the warmer ones. This can be done easily by adjusting the voltage to a higher value for winter service than for summer. The automatic tapering charge still results, but the final rate will be slightly higher. The voltage should not be set so low that there is danger of the voltage of the battery, when fully charged, exceeding the generator voltage in normal operation.

Lamp manufacturers are keenly interested in the application of voltage regulation to automotive requirements, because it assures them of less fluctuation in voltage than formerly and also is good assurance against excessive voltages which endanger lamps. By adjusting the rated voltage of the lamps to the required service-condition, it is being found that lamp life can be greatly prolonged. Battery manufacturers favor voltage regulation because it removes, in a large measure, the chance of constant over-charging of batteries. Energy is furnished to the battery in proportion to its needs, and not at a constant rate which is excessive for a large part of the time.

Aircraft Storage Batteries.—These specifications are intended to apply only to lead-acid storage batteries for aircraft as shown at Fig. 204. Batteries for combined starting and lighting service shall indicate the lighting ability and shall be the capacity in ampere-hours of the battery when it is discharged continuously at the five-hour rate to a final voltage of not less than 1.75 per cell, the temperature of the battery at the beginning of such discharge being 80 degrees Fahrenheit. The second rating shall indicate the starting ability and shall be the minimum current in amperes that the battery will deliver when discharged continuously at the twenty-minute rating to a final voltage of not less than 1.5 per cell, the temperature of the battery at the beginning of such discharge being 80 degrees Fahrenheit.

Means shall be provided to prevent the escape of electrolyte when the battery is turned in an inverted position, or in any position between normal and the inverted position, and allowed to remain in any position for an indefinite period. During this test the electrolyte shall be adjusted to normal level.

Aircraft storage batteries shall be equipped with wing-nut terminals secured to the top of the battery box in such a manner that vibrations from the external leads will be absorbed by the battery box and not transmitted to the terminal post. Both terminals shall be located on the same side of the battery, with the positive to the right when looking at the terminal side of the battery.

QUESTIONS FOR REVIEW

1. How does the Splitdorf SS magneto work?
2. What are the advantages of a double magneto?
3. Describe Delco Battery ignition system.
4. Why is radio shielding necessary and how is it done?
5. Name main features of aircraft engine sparkplugs.
6. What type of generator is best for airplane ignition current supply?

CHAPTER XVI

AIRCRAFT ENGINE LUBRICANTS AND EARLY OILING SYSTEMS

Why Lubrication is Necessary—Friction Defined—Theory of Lubrication—Requirements of Oils—Oil Film Friction—Oil Grooving Bearings—Derivation of Lubricants—Organic Oils—Mineral Lubricants—Properties of Cylinder Oils—Mixed Oils—Flash Test of Oil—Viscosity Measurement—All Oils Contain Carbon—Castor Oil Specifications—Factors Influencing Lubrication System Selection—Pursuit Airplane Engine Lubrication—Gnome Type Engines Use Castor Oil—Hall-Scott Lubrication System—Functions of Bypass Valve—Draining Oil from Crankcase—Oil Supply by Constant Level Splash System—Dry Sump System Best for Airplane Engines—Oiling Curtiss OX Engines—Oil Pumping and Carbon Deposits—Sludge—Rust Corrosion.

The importance of minimizing friction at the various bearing surfaces of machines to secure mechanical efficiency is fully recognized by all mechanics, and proper lubricity of all parts of the mechanism is a very essential factor upon which the durability and successful operation of the motor car powerplant depends. All of the moving members of the engine which are in contact with other portions, whether the motion is continuous or intermittent, of high or low velocity, or of rectilinear or continued rotary nature, should be provided with an adequate supply of oil. No other assemblage of mechanism is operated under conditions which are so much to its disadvantage as the airplane motor and the tendency is toward a development of oiling methods that not only insure that the supply will be ample but will also be automatically applied to the points needing it.

Why Lubrication is Necessary.—In all machinery in motion the members which are in contact have a tendency to stick to each other, and the very minute projections which exist on even the smoothest of surfaces would have a tendency to cling or adhere to each other if the surfaces were not kept apart by some elastic and unctuous substance. This will flow or spread out over the surfaces and smooth out the inequalities existing which tend to produce heat and retard motion of the pieces relative to each other.

A general impression which obtains is that well machined surfaces are smooth, but while they are apparently free from roughness, and no projections are visible to the naked eye, any smooth bearing surface, even if very carefully ground, will have a rough appearance if examined with a magnifying glass. An exaggerated condition to illustrate this point is shown at Fig. 205. The amount of friction will vary in proportion to the pressure on the surfaces in contact and their area and will augment as the loads increase; the rougher surfaces will have more friction than smoother ones and soft bodies will produce more friction than hard substances. Naturally, dry metal will have more friction than lubricated metal.

Friction Defined.—Friction is always present in any mechanism as a resisting force that tends to retard motion and bring all moving parts to a state of rest. The absorption of power by friction may be gauged by the amount of heat which exists at the bearing points. Friction of solids may

be divided into two classes: sliding friction, such as exists between the piston and cylinder, or the bearings of a gas-engine, and rolling friction, which is that present when the load is supported by ball or roller bearings, or that which exists between the tires of the driving wheels and the road. Engineers endeavor to keep friction losses as low as possible, and much care is taken in all modern airplane engines to provide adequate methods of lubrication, or anti-friction bearings at all points where considerable friction exists. When a shaft runs in bearings it may do so under one of three conditions. It may be dry, in which case metallic contact occurs. In that case at high speeds the heat flow due to friction will be excessive, and abrasion will take place. Then there is what is termed boundary friction when the surfaces are coated with a mere smear of oil of little more than molecular thickness. There is little doubt but that in certain parts of an engine—the piston rings for example—a condition of boundary lubrication does exist.

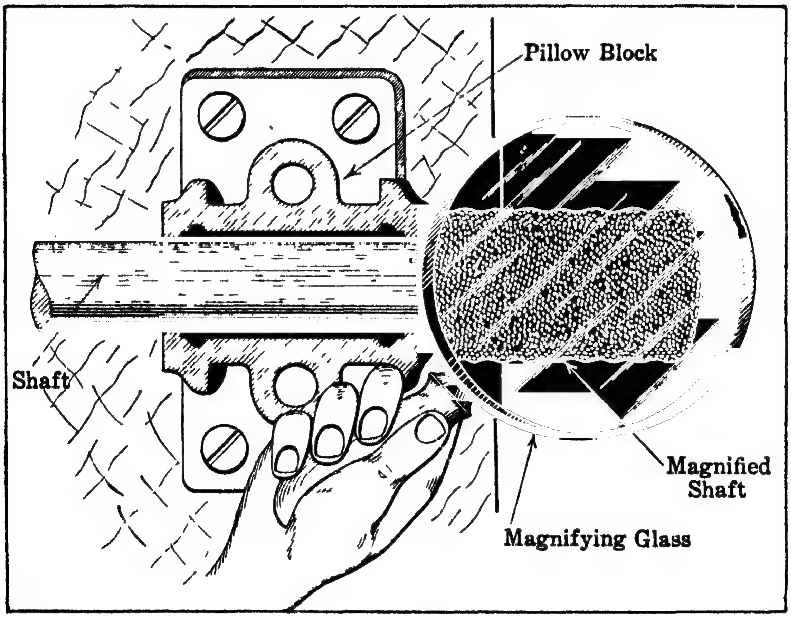


Fig. 205.—Showing Use of Magnifying Glass to Demonstrate that Apparently Smooth Metal Surfaces in Contact Have Minute Irregularities which Produce Friction.

Finally there is the normal condition of flooded lubrication, when the two surfaces are definitely separated by a free oil film. Between boundary and flooded lubrication there is no hard and fast driving line. Flooded conditions may be changed to boundary lubrication as in the case of a journal-bearing where the shaft tends to take up such a position as to wedge the oil film in under the loaded side. This applies almost completely when a free floating bush is introduced between the shaft and bearing, since there is then nothing to restrain the bush from taking up such a position at all times and under all conditions.

A well lubricated bearing will fail when the heat generated by friction exceeds the rate of heat dissipation until the temperature rises enough to melt the bearing metal or vaporize the lubricant. The heat generated is the product of the friction and rubbing speed and nearly proportional to the product of load and rubbing speed. In practice an average connecting-rod big-end bearing can be relied upon to get rid of this heat at a rate to avoid dangerous temperatures. The main bearings of a crankshaft have ample facilities for getting rid of heat by conduction so that they never fail so long as they are lubricated.

Theory of Lubrication.—The reason a lubricant is supplied to bearing points will be easily understood if one considers that these elastic substances flow between the close fitting surfaces, and by filling up the minute depressions in the surfaces and covering the high spots oil films act as cushions which absorb the heat generated and conduct it to the metal backing of the bearing and oil takes the wear instead of the metallic bearing surface. The closer the parts fit together the more fluid the lubricant must be to pass between their surfaces, and at the same time it must possess sufficient body so that it will not be entirely forced out by the pressure existing between the parts.

Oils should have good adhesive, as well as cohesive, qualities. The former are necessary so that the oil film will cling well to the surfaces of the bearings; the latter, so the oil particles will cling together and resist the tendency to separation which exists all the time the bearings are in operation. When used for gas-engine lubrication the oil should be capable of withstanding considerable heat in order that it will not be vaporized by the hot portions of the cylinder. It should have sufficient cold test so that it will remain fluid and flow readily at low temperature. Lubricants should be free from acid, or alkalies, which tend to produce a chemical action with metals and result in corrosion of the parts to which they are applied. It is imperative that the oil be exactly the proper quality and nature for the purpose intended and that it be applied in a positive manner.

Requirements of Oils.—The requirements may be briefly summarized as follows:

First—It must have sufficient body to prevent seizing of the parts to which it is applied and between which it is depended upon to maintain an elastic film, and yet it must not have too much viscosity, in order to minimize the internal or fluid friction which exists between the particles of the lubricant itself and which might prevent free flow.

Second—The lubricant must not coagulate or gum; must not injure the parts to which it is applied, either by chemical action or by producing injurious deposits, and it should not evaporate readily.

Third—The character of the work will demand that the oil should not vaporize when heated or thicken to such a point that it will not flow readily when cold.

Fourth—The oil must be free from acid, alkalies, animal or vegetable fillers, or other injurious agencies.

Fifth—It must be carefully selected for the work required and should be a good conductor of heat.

Oil Film Friction.—Albert Kingsbury's experiments on the lubrication of journal-bearings, with special reference to the prevention of wear through the separation of the surfaces by proper construction and oiling are considered a classic by mechanical engineers and the writer cannot do better than mention some of the results obtained by this indefatigable investigator. During his experiments at Cornell University as early as 1888, Mr. Kingsbury, by skillfully fitting with equal care various bronze bearings to the journal of the testing-machine, was able to measure the friction of the oil alone; and it was found to be the same with all bearings. By measuring the air-film thicknesses, the pressures and the friction all round an air-lubricated journal-bearing, he then confirmed the correctness of Reynolds' mathematical explanation of lubrication. To him belongs the credit of obtaining the lowest reliable experimentally determined coefficient of friction, for an oil-lubricated journal, namely, 0.00053. Other researches showed the oil viscosity to be a direct factor in the friction of journal-bearings, being inversely proportional to the film thickness down to the least thickness that could be measured, which was about 0.000025 inch.

Bearings are classified and an equation is given for the factors that influence the friction of an oil-film, which is of calculable amount. Neglecting gravity, no pressure is said to be present in the oil-film so long as the journal and the bearing are côncentric, but the greater the eccentricity, the greater will be the pressure. Theoretically, it is impossible to make the journal touch the bearing while the journal rotates and an oil-film fills the clearance space. But just as the pressures in the oil-film increase with the eccentricity, so also increases the frictional resistance to rotation of the journal.

The reader should consider the following classification of journal-bearings, all of which are used in aviation engines:

(1) Rotation
- (*a*) Continuous
  - (1) in one direction
  - (2) in either direction
- (*b*) Oscillating
  - (1) wide angle
  - (2) small angle

(2) Speed
- (*a*) High
- (*b*) Medium
- (*c*) Low

(3) Pressure
- (*a*) Unidirectional
- (*b*) Reversible

(4) Lubrication
- (*a*) Copious and continuous
- (*b*) Scanty and intermittent

(5) Design
- (*a*) Long or short
- (*b*) Full or partial
- (*c*) Clearance or fitted

(*d*) Plain or grooved

Let us first consider a short bearing, one journal diameter long, bored with a running clearance, η, in which the journal runs at moderate speed in one direction only to show how oil film friction may be measured. There is no groove in the bearing except the oil-distributing channel at the top. The oil flows down from this as fast as it leaks away at the bottom.

If the shaft is vertical and so balanced that there is no force to move it sidewise, it will run exactly in the center of the bearing. Neglecting gravity, there will be no pressure within this oil-film so long as the journal and the bearing are concentric. There will be friction of a calculable amount, however. Its moment, resisting rotation, will be

$$M = (1.65\,\mu\, d^2\, l\, N) / (2\eta / d) \text{ in-lb.} \qquad (1)$$

In this equation, we find the factors that influence the friction

μ = coefficient of viscosity of the oil, in pounds, inches, seconds

The more viscous the oil, the greater will be the friction.

η = the radial clearance, in inches

The greater the clearance, the less will be the friction.

N = revolutions per minute

The higher the speed, the greater will be the friction.

d = diameter of journal, in inches

The larger the diameter, the greater will be the friction.

l = length of bearing, in inches

The shorter the length, the less will be the friction.

Obviously, the shaft might run in closer proximity to one side so that the least film-thickness would be 0.5η, and the greatest 1.5η. The axial eccentricity would be 0.5η, or 50 per cent of the radial clearance. This percentage of eccentricity is designated by the letter c. The journal will not run eccentrically, however, unless it is pushed over. When pushed out of center, the journal meets opposition from pressures that then arise in the oil-film.

To Mr. Kingsbury belongs the credit for obtaining the lowest reliable experimentally determined coefficient of friction. In his paper, A New Oil-Testing Machine, published in the *Transactions of the A. S. M. E.*, he reported a coefficient of friction of 0.00053, but later he obtained 0.00046 in some other tests that have never been published. These coefficients were for an oil-lubricated journal, the bearing of which was fitted for about 120 degrees. These low values were the result of very careful workmanship. They were far below the lowest coefficient published for ball-bearings up to that time. In this connection it is only fair to state that while extremely low coefficients of friction have been obtained under ideal laboratory conditions for plain journals, in practical application and with existing shop methods the friction of ball-bearings is less than that of a plain bearing, no matter how well lubricated it is, under similar conditions of speed and load. Then again, one must consider that the friction of a ball-bearing is practically independent of lubrication, while that of a plain journal is materially influenced by the degree and character of the lubricating medium.

Oil-Grooving Bearings.—Some authorities state there should be no oil-grooves in the loaded side of a bearing, because oil-grooves are channels

through which the oil-pressures in the film are relieved. This relieving of the pressure thins the film and makes the bearing run hotter than it would otherwise.

A few general rules for grooving may be followed with safety:

(1) When the bearing completely surrounds the journal, the bearing-surface may be unbroken by grooves, except one longitudinal groove along the surface about opposite to the loaded side. For low speeds, the single groove may be small with closed ends, with drip lubrication, or, for high speed, the groove may be large, through which cool oil may be circulated rapidly to remove the heat of friction.

(2) Partial bearings are used on railroads but are seldom found in engine construction. They are used with the bearings above the journal, and as lineshaft bearings in ships, when the bearing is below the journal. The circumferential length of a partial bearing is usually 120 degrees, or less. It can then be bored with a clearance, the amount of which depends upon the load and the lubrication, or it can be fitted carefully to the journal and be suitable for heavy service. If a partial bearing covers more than 120 degrees of the journal, it should always be provided with running clearance. Partial bearing surfaces require no grooves. The leading edge, where oil enters the film, should be slightly rounded.

(3) Full bearings, with two opposite grooves, about 90 degrees from the loaded side, are very common, in bearings made in halves. No additional grooves should be used. Such bearings should be bored with running clearance, the amount depending on the speed and the load. Heavily loaded bearings may require some fitting even though provided with running clearance.

Derivation of Lubricants.—The first oils which were used for lubricating machinery were obtained from animal and vegetable sources, though at the present time most unguents are of mineral derivation. Lubricants may exist as fluids, semifluids, or solids. The viscosity will vary from light spindle or dynamo oils, which have but little more body than kerosene, to the heaviest greases and tallows. The most common solid employed as a lubricant is graphite, sometimes termed "plumbago" or "black lead." This substance is of mineral derivation. The disadvantage of oils of organic origin, such as those obtained from animal fats or vegetable substances, is that they will absorb oxygen from the atmosphere, which causes them to thicken or become rancid. Such oils have a very poor cold test, as they solidify at comparatively high temperatures, and their flashing point is so low that they cannot be used at points where much heat exists. In most animal oils various acids are present in greater or less quantities, and for this reason they are not well adapted for lubricating metallic surfaces which may be raised high enough in temperature to cause decomposition of the oils and acid reaction with the metals lubricated. Organic oils also decompose by heat and produce gummy deposits in the combustion-chamber.

Lubricants derived from the crude petroleum are called "Oleonaphthas" and they are a product of the process of refining petroleum through which

gasoline and kerosene are obtained. They are of lower cost than vegetable or animal oil, and as they are of nonorganic origin, they do not become rancid or gummy by constant exposure to the air, and they will have no corrosive action on metals because they contain no deleterious substances in chemical composition. By the process of fractional distillation mineral oils of all grades can be obtained. They have a lower cold and higher flash test and there is not the liability of spontaneous combustion that exists with animal oils.

Organic Oils.—The organic oils are derived from fatty substances, which are present in the bodies of all animals and in some portions of plants. The general method of extracting oil from animal bodies is by a rendering process, which consists of applying sufficient heat to liquefy the oil and then separating it from the tissue with which it is combined by compression. The only oil which is used to any extent in gas-engine lubrication that is not of mineral derivation is castor oil. This substance has been used on high-speed racing automobile engines and on airplane powerplants. It is obtained from the seeds of the castor plant, which contain a large percentage of oil.

Among the solid substances which have been used for lubricating purposes in machinery may be mentioned tallow, which is obtained from the fat of animals, and graphite and soapstone, which are of mineral derivation. Tallow is never used at points where it will be exposed to much heat, though it was sometimes employed as a filler for greases used in transmission gearing of autos and on machinery bearings.

Mineral Lubricants.—Graphite is sometimes mixed with oil and applied to cylinder lubrication, though it is most often used in connection with greases in the landing gear parts and for coating wires and cables of the airplane. Graphite is not affected by heat, cold, acids, or alkalies, and has a strong attraction for metal surfaces. It mixes readily with oils and greases and increases their efficiency in many applications. It is sometimes used where it would not be possible to use other lubricants because of extremes of temperature as in exhaust valve guides.

The oils used for cylinder lubrication are obtained almost exclusively from crude petroleum derived from American wells. Special care must be taken in the selection of crude material, as every variety will not yield oil of the proper quality to be used as a cylinder lubricant. The crude petroleum is distilled as rapidly as possible with fire heat to vaporize off the naphthas and the burning oils. After these vapors have been given off superheated steam is provided to assist in distilling. When enough of the light elements have been eliminated the residue is drawn off, passed through a strainer to free it from grit and earthy matters, and is afterwards cooled to separate the wax from it. This is the dark cylinder oil and is the grade usually used for steam-engine cylinders.

Properties of Cylinder Oils.—The oil that is to be used in the gasoline-engine must be of high quality, and for that reason the best grades are distilled in a vacuum that the light distillates may be separated at much lower temperatures than ordinary conditions of distilling permit. If the degree of heat is not high the product is not so apt to decompose and deposit carbon. If it is desired to remove the color of the oil which is caused by free

carbon and other impurities it can be accomplished by filtering the oil through charcoal. The greater the number of times the oil is filtered, the lighter it will become in color. The best cylinder oils have flash points usually in excess of 500 degrees Fahrenheit, and while they have a high degree of viscosity at 100 degrees Fahrenheit they become more fluid as the temperature increases.

The lubricating oils obtained by refining crude petroleum may be divided into three classes:

First—The natural oils of great body which are prepared for use by allowing the crude material to settle in tanks at high temperature and from which the impurities are removed by natural filtration. These oils are given the necessary body and are freed from the volatile substances they contain by means of superheated steam which provides a source of heat.

Second—Another grade of these natural oils which are filtered again at high temperatures and under pressure through beds of animal charcoal to improve their color.

Third—Pale, limpid oils, obtained by distillation and subsequent chemical treatment from the residuum produced in refining petroleum to obtain the fuel oils.

Mixed Oils.—Authorities agree that any form of mixed oil in which animal and mineral lubricants are combined should never be used in the cylinder of a gas-engine of any type using fixed cylinders as the admixture of the lubricants does not prevent the decomposition of the organic oil into the glycerides and fatty acids peculiar to the fat used. In a gas-engine cylinder the flame tends to produce more or less charring. The deposits of carbon will be much greater with animal oils than with those derived from the petroleum base because the constituents of a fat or tallow are not of the same volatile character as those which comprise the hydro-carbon oils which will evaporate or volatilize before they char in most instances. This does not apply with equal force to the use of blended mineral and castor oil or to certain types of aviation engines such as the early Gnome and Le Rhone rotary engines which used castor oil because of the method of fuel vapor charge distribution to the cylinders.

All lubrication experts claim that pure petroleum products are the best oils for the lubrication of internal-combustion engines and that the quality of these oils is largely determined by the degree of purity to which they are brought in the refining process. For the lubrication of the early French rotary type motors where the lubricating oil comes in direct contact with the fuel, it has been found that vegetable castor oil is better, because it is less soluble in gasoline and will, therefore, stick better to the surfaces which are washed by the fuel. Such engines are seldom used at the present time because of the marked development of the fixed radial types.

European engineers were almost forced to the use of oils of vegetable and animal origin during the War owing to the fact that Europe has no available supply of high grade petroleum, and of all such lubricants, pure castor oil is unquestionably the best. In fact, under certain conditions it makes a very satisfactory lubricant. The objections to its use are, that being a vegetable product, it contains a considerable amount of gums and resins which accumulate in the engine and collect on the valves and piston

rings and that in heating and cooling it absorbs oxygen which gives it a tendency to become too thick and gummy. In rotary engines, any excess oil was thrown out of the exhaust by centrifugal force and early pilots flying training ships such as the Nieuports and Avros or service ships such as the Camel know the taste and smell of burnt castor oil very well.

In some instances a mixture of castor and mineral oils is used, but exhaustive tests made by the U. S. Army Air Service has proved that such a mixture has no advantage over a straight mineral oil, except in the case of rotary engines. In making a mixture of castor and mineral oil, the two not being mutually soluble, a mutual solvent such as lard oil or Neatsfoot oil which are of animal origin must be used, but all such mutual solvents have some undesirable characteristics of their own.

No laboratory tests have yet been devised which will determine absolutely whether or not a given oil will give satisfactory service in an engine and it is the usual practice among the reliable refiners of oil to first make an oil having the desired chemical characteristics and then subject that oil to severe and exhaustive tests in an engine. Such tests are usually made on a test stand or a dynamometer where operating conditions may be kept constant during the tests and where measurements of temperature, fuel consumption, horsepower, etc., may be made conveniently and accurately.

Flash Test of Oil.—The flash test reveals the lowest temperature at which an oil will give off a combustible vapor at atmospheric pressure in the presence of air. The test is made by carefully heating a sample of the oil and noting the temperature at which the vapors first ignite. The flash point of any oil is practically the flash point of its lightest constituent. When an oil, therefore, has a high flash point, that fact in itself is evidence that it is not a mixture of light and heavy oils of different composition. It is apparent that a few drops of gasoline, due to excessive priming, or a slight amount of dilution with lighter oils will cause the flash point to drop immediately. U. S. Government specifications require a flash point of not less than 400 degrees Fahrenheit for light aero oils and not less than 450 degrees Fahrenheit for heavy aero oils. The fire point is the temperature at which the vapors given off by heating the oil will ignite and continue to burn. It is from 40 to 60 Fahrenheit degrees above the flash point. Government specifications do not cover the fire point, as it is of little value when the flash point of the oil is known.

Viscosity Measurement.—Viscosity, as applied to aero oils, represents the resistance to flow and is measured by taking the time, in seconds, required for a given amount of oil to flow through a standard nozzle at a given temperature. All oils become thin as they are heated, some of them at a much faster rate than others, hence, in order for the test to be of any value, the exact temperature at which the viscosity is taken must be known. In order to know the loss of viscosity when oil is heated, it is necessary to take viscosity readings at a number of different temperatures.

Government specifications require the following viscosities at 210 degrees Fahrenheit.

Grade 1	75 to 85 seconds
Grades 2 and 3	90 to 100 seconds
Grade 4	115 to 125 seconds

Government specifications also call for a cold test which simply means that oils must pour or flow at certain temperatures. The grades are:

Grade 1	15° F.
Grade 2	30° F.
Grades 3 and 4	45° F.

The Grade 1 oil is used for winter flying in colder climates and for flights at altitudes where low temperatures are encountered. Even when flying under these conditions it is doubtful if a temperature lower than fifteen degrees Fahrenheit would ever be encountered in the lubricating system and the oil system should always be thoroughly drained immediately following each flight.

All Oils Contain Carbon.—All petroleum oils have over 80 per cent carbon by weight and good lubricating oils consist exclusively of carbon and hydrogen combined in definite chemical relation. The carbon residue is found by a test in which the oil is subjected to destructive distillation known as the Conradson test. The carbon residue does not indicate, in any manner, the rapidity with which carbon deposits may be expected to form in the cylinders and on the pistons of an engine. Accumulation of carbon in an aircraft engine is due, in practically every instance, to imperfect combustion of the fuel and even in cases of excessive oil consumption or oil pumping, stoppage is generally caused by fouling of the sparkplugs rather than to excessive carbon residue from the oil. It is true that carbon will accumulate in any engine operated for a sufficient length of time, but the use of a proper fuel mixture and tight fitting piston rings will do more to minimize this condition than any slight change in the carbon residue of the oil possibly could.

Government specifications allow a maximum carbon residue of 1.5 per cent for Grade 1, 2.0 per cent for Grade 2 and 2.5 per cent for Grades 3 and 4. The carbon residue is taken as an index to the degree of refining and filtration employed in the manufacture of the oil.

U. S. Government specifications for aero oil also require detailed laboratory tests to determine acidity, precipitation of insoluble matter and demulsibility, all of which are of a chemical nature and are designed to show the degree of refinement and filtration to which the oil has been subjected in its manufacture. None of them has any direct bearing on the use of an oil in an engine and none gives any indication as to the lubricating properties of the oil. The leading oil refiners have selected certain grades and blends which they know are suitable for aero engines and the recommendations of a reliable oil manufacturer can usually be followed to advantage. All engine builders recommend grades of oil best adapted to their product and the user will not go wrong if he follows their advice.

Castor Oil Specification.—The U. S. Air Service Specification No. 3,500 B covers the desired characteristics of castor oil for aircraft engine lubrication. The oil must be a high-grade vegetable castor oil suitable for rotary engine lubrication. Both cold-pressed vegetable castor oil and hot-pressed vegetable castor oil which has been refined so that it will meet the requirements of this specification are acceptable. The castor oil must be free from adulteration, other oils, suspended matter, grit and water.

Physical Properties and Tests.—The castor oil must meet the following requirements:

Color—When observed in a four-ounce sample bottle, the castor oil must be colorless or nearly so, transparent, and without fluorescence.

Specific Gravity—The castor oil must have a specific gravity of 0.959 to 0.968 at 60 degrees Fahrenheit. (Baumé gravity must be from 16.05 to 14.70 at 60 degrees Fahrenheit.)

Viscosity—The castor oil, when tested in a Saybolt universal viscosimeter, must have a viscosity of not less than 450 seconds at 130 degrees Fahrenheit and 95 seconds at 212 degrees Fahrenheit.

Flash Point—The flash point must not be less than 450 degrees Fahrenheit in a Cleveland open cup flash tester.

Pour Test—The castor oil, in a four-ounce sample bottle one-quarter (1/4) full, must not congeal on being subjected to a temperature of plus five degrees Fahrenheit for one hour. (See A. S. Specification No. 3,525 for "Pour-test.")

Evaporation Test—The castor oil must not show a greater loss than five-tenths (0.5) of one per cent when heated in an oven at 230 degrees Fahrenheit for one and three-quarter hours. This test shall be made on a five-gram sample in a glass crystallizing dish approximately 2 1/2 inches in diameter and 1 1/8 inches high, inside dimensions.

Ash—The castor oil shall not show more than 0.015 per cent ash and shall show no impurity of any sort not related to the original product.

Chemical Properties and Tests.—Solubility—The castor oil must be completely soluble in four volumes of 90 per cent alcohol (specific gravity 0.834 at 60 degrees Fahrenheit). This test shall be made on a two cubic centimeter sample.

Acid Number—It must not require more than three milligrams of potassium hydroxide (KOH) or 2.14 milligrams of sodium hydroxide ($NaOH$) to neutralize one gram of oil. This is equivalent to 1.5 per cent oleic acid.

Unsaponifiable Matter—The unsaponifiable matter must not exceed one per cent. Samples used for this test shall weigh five to ten grams.

Iodine Number (Hanus or Wijs methods)—The iodine number must be between 80 and 90. Samples used for this test shall weigh 0.2 to 0.25 gram and shall be treated for one hour.

Rosin (Lieberman-Storch test)—The castor oil must not give a reaction for either rosin or rosin oil.

Cottonseed Oil (Halphen test)—The castor oil must not give a reaction for cottonseed oil.

Factors Influencing Lubrication System Selection.—The suitability of oil for the proper and efficient lubrication of all internal-combustion engines is determined chiefly by the following factors:

1. Type of cooling system (operating temperatures).
2. Type of lubricating system (method of applying oil to the moving parts).
3. Rubbing speeds of contact surfaces.

Were the operating temperatures, bearing surface speeds and lubrication systems identical, a single oil could be used in all engines with equal satisfaction. The only change then necessary in viscosity would be that

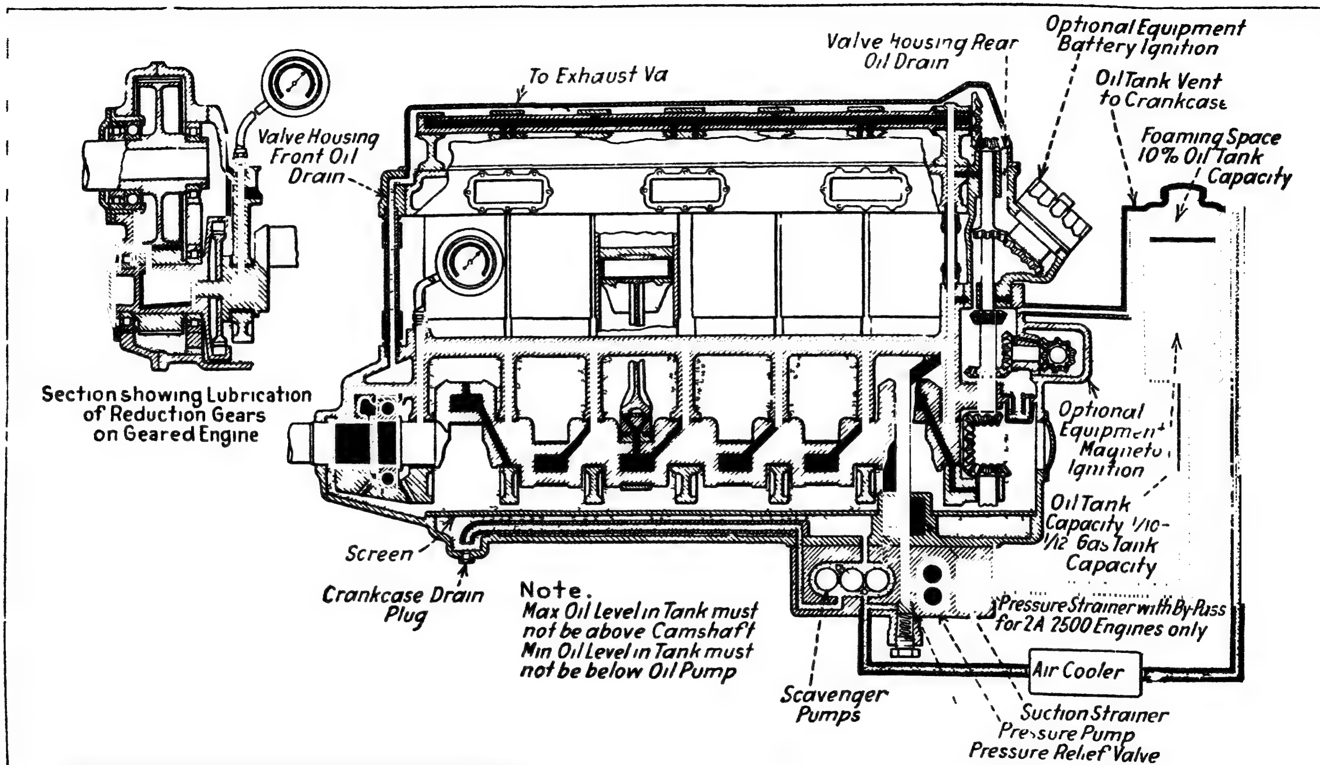


Fig. 206.—Oiling System of Packard Aircraft Engines is Typical of Latest Practice in Lubrication of Motor Parts. Two Pumps are Used, One to Force Oil to All Bearing Points, the Other to Draw it from the Sump at the Bottom of the Crankcase.

due to climatic conditions. As engines are now designed, only three grades of oil are necessary for the lubrication of all types with the exception of Knight, air-cooled and some engines which run continuously at full load. In the specification of engine lubricants the feature of load carried by the engine should be carefully considered.

Full Load Engines.
1. Marine.
2. Racing automobile.
3. Aviation.
4. Farm tractor.
5. Some stationary.

Variable Load Engines.
1. Passenger automobile.
2. Commercial vehicle.
3. Motor cycle.
4. Some stationary.

Of the forms outlined, the only one we have any immediate concern about is the airplane powerplant. The Platt & Washburn Refining Company, who have made a careful study of the lubrication problem as applied to all types of engines, have found a peculiar set of conditions to apply to oiling high-speed constant-duty or "full-load" engines. Modern airplane engines are designed to operate continuously at a fairly uniform high rotative speed and at full load over long periods of time. As a sequence to this heavy duty the operating temperatures are elevated. For the sake of extreme lightness in weight of all parts, very thin alloy steel, aluminum or cast iron pistons are fitted and the temperature of the thin piston heads at the center reaches anywhere between 600 degrees and 1,400 degrees Fahrenheit, as in automobile racing engines. Freely exposed to such intense heat hydrocarbon oils are partially "cracked" into light and heavy products or polymerized into solid hydrocarbons. From these facts it follows that only heavy mineral oils of low carbon residue and of the greatest chemical purity and stability should be used to secure good lubrication. In all cases the oil should be sufficiently heavy to assure the highest horsepower and fuel and oil economy compatible with perfect lubrication, avoiding, at the same time, carbonization and ignition failure. When aluminum pistons are used their superior heat-conducting properties aid materially in reducing the rate of oil destruction.

Pursuit Airplane Engine Lubrication.—The extraordinary evolutions described by military airplanes in flight make it a matter of vital necessity to operate engines inclined at all angles to the vertical as well as in an upside-down position. To meet this situation lubricating systems have been elaborated so as to deliver an abundance of oil where needed and to eliminate possible flooding of cylinders. This is done by applying a full force feed system, distributing oil under considerable pressure to all working parts. Discharged through the bearings, the oil drains down to the suction side of a second pump located in the bottom of the base chamber. This pump being of greater capacity than the first prevents the accumulation of oil in the crankcase, and forces it to a separate oil reservoir-cooler, whence it flows back in rapid circulation to the pump feeding the bearings. With this arrangement positive lubrication is entirely independent of engine position The lubricating system of the Packard aviation engines, which is shown at Fig. 206, is typical of current practice. The lubricating method of the Curtiss D12 engine shown at Fig. 207 is also a good example of a dry sump

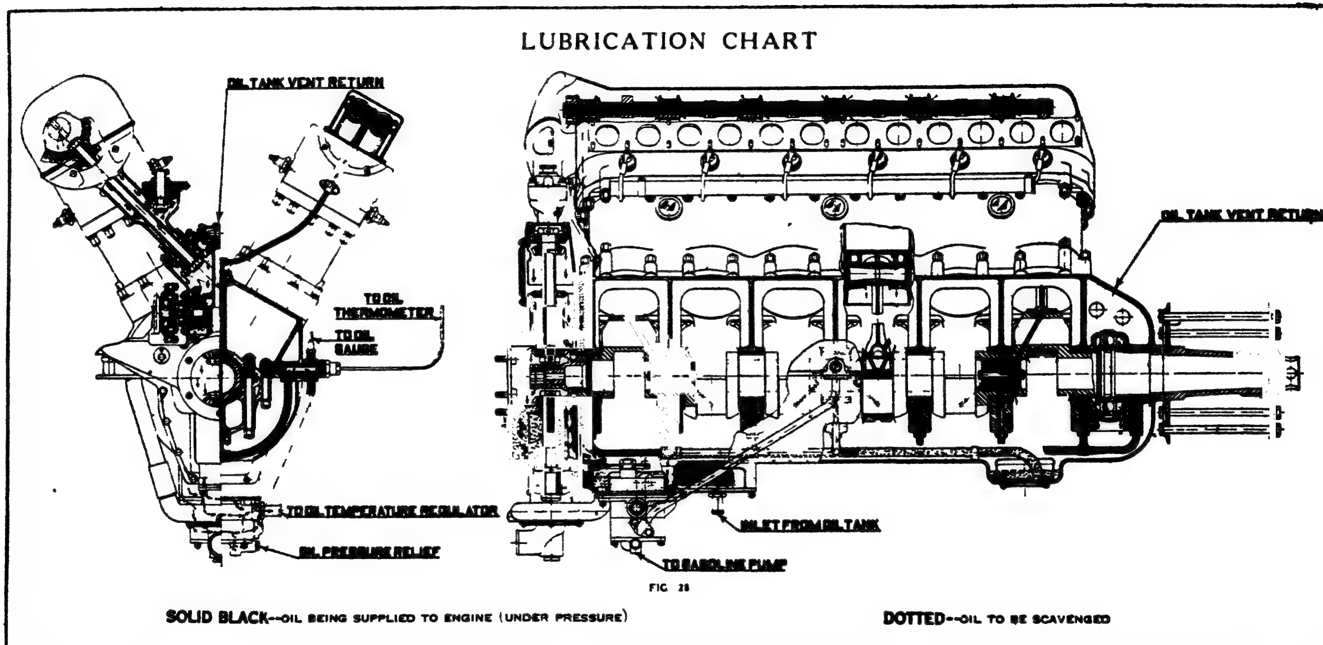


Fig. 207.—Lubrication Chart of the Curtiss D12 Aviation Engine, Showing Method of Supplying Oil Under Pressure and Use of the Pump to Keep the Crankcase Drained.

system applied to a water-cooled engine. The Wright Whirlwind Model J5 engine uses the oiling system shown at Fig. 208 and this may be considered typical of air-cooled radial engines. All of these systems will be described in detail in other chapters.

Gnome Type Engines Use Castor Oil.—The construction and operation of rotative radial cylinder engines introduce additional difficulties of lubrication to those already referred to and merit especial attention. Owing to the peculiar fuel supply systems of Gnome type engines in which atomized gasoline mixed with air is drawn through the hollow stationary crankshaft directly into the crankcase, which it fills on the way to the cylinders, ordinary mineral oils cannot be used. Hydrocarbon oils are soon dissolved by the gasoline and washed off, leaving the bearing surfaces without adequate protection and exposed to instant wear and destruction. So castor oil is resorted to as an indispensable but unfortunate compromise. Of vegetable origin, it leaves a much more bulky carbon deposit in the explosion chambers than does mineral oil and its great affinity for oxygen causes the formation of voluminous gummy deposit in the crankcase. Engines employing it need to be dismounted and thoroughly scraped out at frequent intervals. It is advisable to use only unblended chemically pure castor oil in rotative engines, first by virtue of its insolubility in gasoline and second because its extra heavy body can resist the high temperature of air-cooled cylinders though this advantage is also shared by the heavy bodied mineral oils.

Hall-Scott Lubrication System.—The oiling system of the Hall-Scott type A5 125 horsepower water-cooled engine is clearly shown at Fig. 209 and may be considered typical of prewar practice. It was completely described in the instruction book issued by the company from which the following extracts are reproduced. Crankshaft, connecting rods and all other parts within the crankcase and cylinders are lubricated directly or indirectly by a force-feed oiling system. The cylinder walls and wristpins are lubricated by oil spray thrown from the lower end of connecting rod bearings. This system is used only upon A5 engines. Upon A7a and A5a engines a small tube supplies oil from connecting rod bearing directly upon the wristpin. The oil is drawn from the strainer located at the lowest portion of the lower crankcase, forced around the main intake manifold oil jacket. From here it is circulated to the main distributing pipe located along the lower left hand side of upper crankcase. The oil is then forced directly to the lower side of crankshaft, through holes drilled in each main bearing cup. Leakage from these main bearings is caught in scuppers placed upon the cheeks of the crankshafts furnishing oil under pressure to the connecting rod bearings. A7a and A5a engines have small tubes leading from these bearings which convey the oil under pressure to the wristpins.

Function of Bypass Valve.—A bypass located at the front end of the distributing oil pipe can be regulated to lessen or raise the pressure. By screwing the valve in, the pressure will rise and more oil will be forced to the bearings. By unscrewing, pressure is reduced and less oil is fed. A7a and A5a engines have oil relief valves located just off of the main oil pump in the lower crankcase. This regulates the pressure at all times so that in cold weather there will be no danger of bursting oil pipes due to

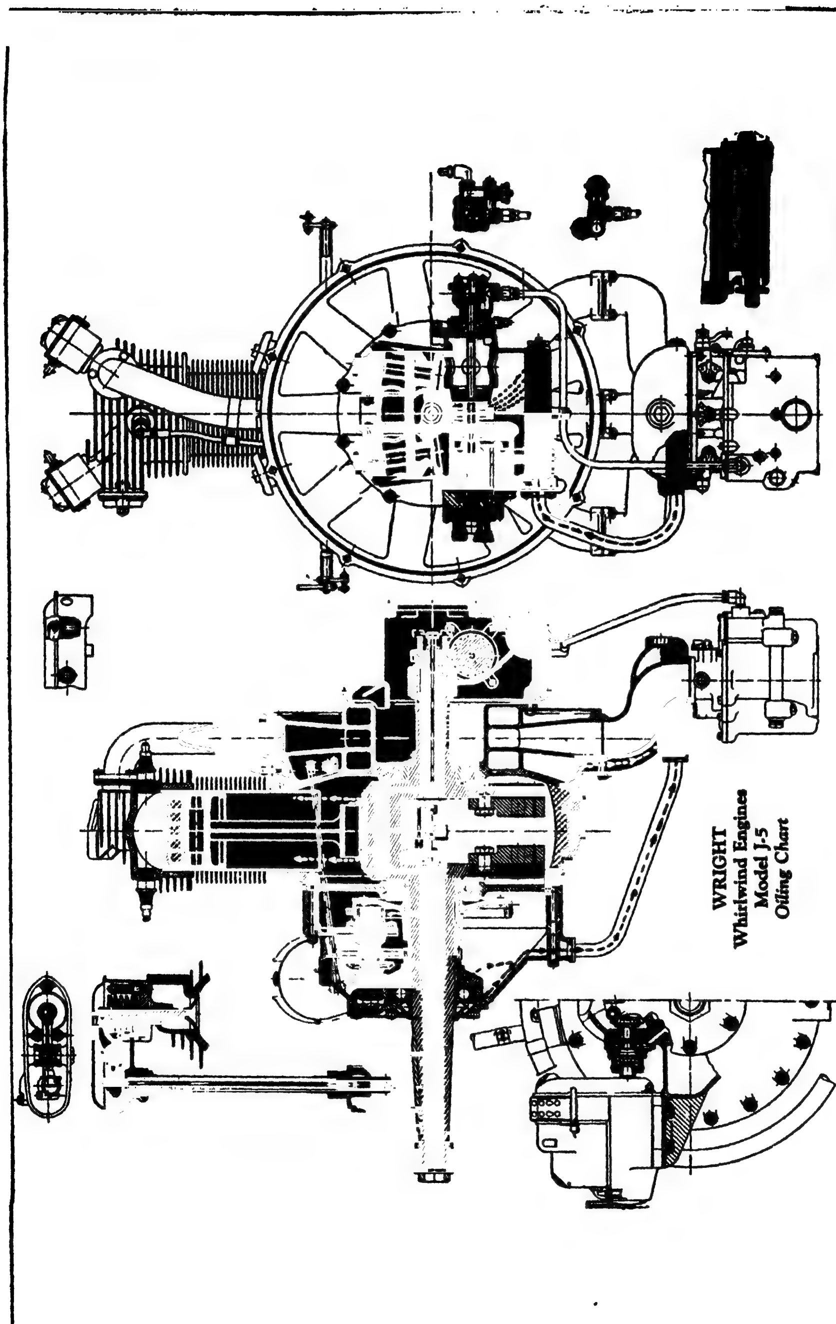


Fig. 208.—Oiling Diagram of Wright "Whirlwind" Model J5 Engines.

excessive pressure. If it is found the oil pressure is not maintained at a high enough level, inspect this valve. A stronger spring will not allow the oil to bypass so freely, and consequently the pressure will be raised; a weaker spring will bypass more oil and reduce the oil pressure materially.

Independent of the above-mentioned system, a small, directly driven rotary oiler feeds oil to the base of each individual cylinder. The supply of oil is furnished by the main oil pump located in the lower crankcase. A small sight-feed regulator is furnished to control the supply of oil from this oiler. This instrument should be placed higher than the auxiliary oil distributor itself to enable the oil to drain by gravity feed to the oiler. If there is no available place with the necessary height in the front seat of plane, connect it directly to the intake L fitting on the oiler in an upright position. It should be regulated with full open throttle to maintain an oil level in the glass, approximately half way.

An oil pressure gauge is provided. This should be run to the pilot's instrument board. The gauge registers the oil pressure upon the bearings, also determining its circulation. Strict watch should be maintained of this instrument by pilot, and if for any reason its hand should drop to O the motor should be immediately stopped and the trouble found before restarting engine. Care should be taken that the oil does not work up into the gauge, as it will prevent the correct gauge registering of oil pressure. The oil pressure will vary according to weather conditions and viscosity of oil used. In normal weather, with the engine properly warmed up, the pressure will register on the oil gauge from five to ten pounds when the engine is turning from 1,275 to 1,300 r.p.m. This does not apply to all aviation engines, however, as the proper pressure advised for the Curtiss OX2 motor is from 40 to 55 pounds at the gauge. As a rule, engines using wet sump lubrication employ lower pressure than those lubricated by the dry sump method.

The oil sump plug is located at the lowest point of the lower crankcase. This is a combination dirt, water and sediment trap. It is easily removed by unscrewing. Oil is furnished mechanically to the camshaft housing under pressure through a small tube leading from the main distributing pipe at the propeller end of engine directly into the end of camshaft housing. The opposite end of this housing is amply relieved to allow the oil to rapidly flow down upon camshaft, magneto, pinionshaft, and crankshaft gears, after which it returns to lower crankcase. An outside overflow pipe is also provided to carry away the surplus oil.

Draining Oil from Crankcase.—The oil strainer is placed at the lowest point of the lower crankcase. This strainer should be removed after every five to eight hours running of the engine and cleaned thoroughly with gasoline. It is also advisable to squirt distillate up into the case through the opening where the strainer has been removed. Allow this distillate to drain out thoroughly before replacing the plug with strainer attached. Be sure gasket is in place on plug before replacing. Pour new oil in through either of the two breather pipes on exhaust side of motor. Be sure to replace strainer screens if removed. If, through oversight, the engine does not receive sufficient lubrication and begins to heat or pound, it should be stopped immediately. After allowing engine to cool pour at least three

gallons of oil into oil sump. Fill radiator with water after engine has cooled. Should there be apparent damage, the engine should be thoroughly inspected immediately without further running. If no obvious damage has been done, the engine should be given a careful examination at the earliest opportunity to see that the running without oil has not burned the bearings or caused other trouble. Oils best adapted for Hall-Scott engines have the following properties: A flash test of not less than 400 degrees Fahrenheit; viscosity of not less than 75 to 85 taken at twenty degrees Fahrenheit with Saybolt's Universal Viscosimeter.

Zeroline heavy duty oil, manufactured by the Standard Oil Company of California; also,

Gargoyle mobile B oil, manufactured by the Vacuum Oil Company, both fulfill the above specifications. One or the other of these oils can be obtained all over the world. Monogram extra heavy is also recommended.

Oil Supply by Constant Level Splash System.—The splash system of lubrication that depends on the connecting rod to distribute the lubricant is one of the most successful and simplest forms for simple four- and six-cylinder vertical automobile engines, but is not as well adapted to the oiling of airplane powerplants for reasons previously stated. If too much oil is supplied the surplus will work past the piston rings and into the combustion-chamber, where it will burn and cause carbon deposits. Too much oil will also cause an engine to smoke and an excess of lubricating oil is usually manifested by a bluish-white smoke issuing from the exhaust.

A good method of maintaining a constant level of oil for the successful application of the splash system is shown at Fig. 210. The engine base casting includes a separate chamber which serves as an oil container and which is below the level of oil in the crankcase. The lubricant is drawn from the sump or oil container by means of a positive oil pump which discharges directly into the engine case. The level is maintained by an overflow pipe which allows all excess lubricant to flow back into the oil container at the bottom of the cylinder. Before passing into the pump again the oil is strained or filtered by a screen of wire gauze and all foreign matter removed. Owing to the rapid circulation of the oil it may be used over and over again for quite a period of time. The oil is introduced directly into the crankcase by a breather pipe and the level is indicated by a rod carried by a float which rises when the container is replenished and falls when the available supply diminishes. It will be noted that with such system the only apparatus required besides the oil tank which is cast integral with the bottom of the crankcase is a suitable pump to maintain circulation of oil. This member is always positively driven, either by means of shaft and universal coupling or direct gearing. As the system is entirely automatic in action, it will furnish a positive supply of oil at all desired points, and it cannot be tampered with by the inexpert because no adjustments are provided or needed.

Dry Sump System Best for Airplane Engines.—In most airplane powerplants it is considered desirable to supply the oil directly to the parts needing it by suitable leads instead of depending solely upon the distributing action of scoops on the connecting rod big ends. Systems of this nature are shown at Figs. 206, 207 and 208. The oil is carried in the crankcase, as

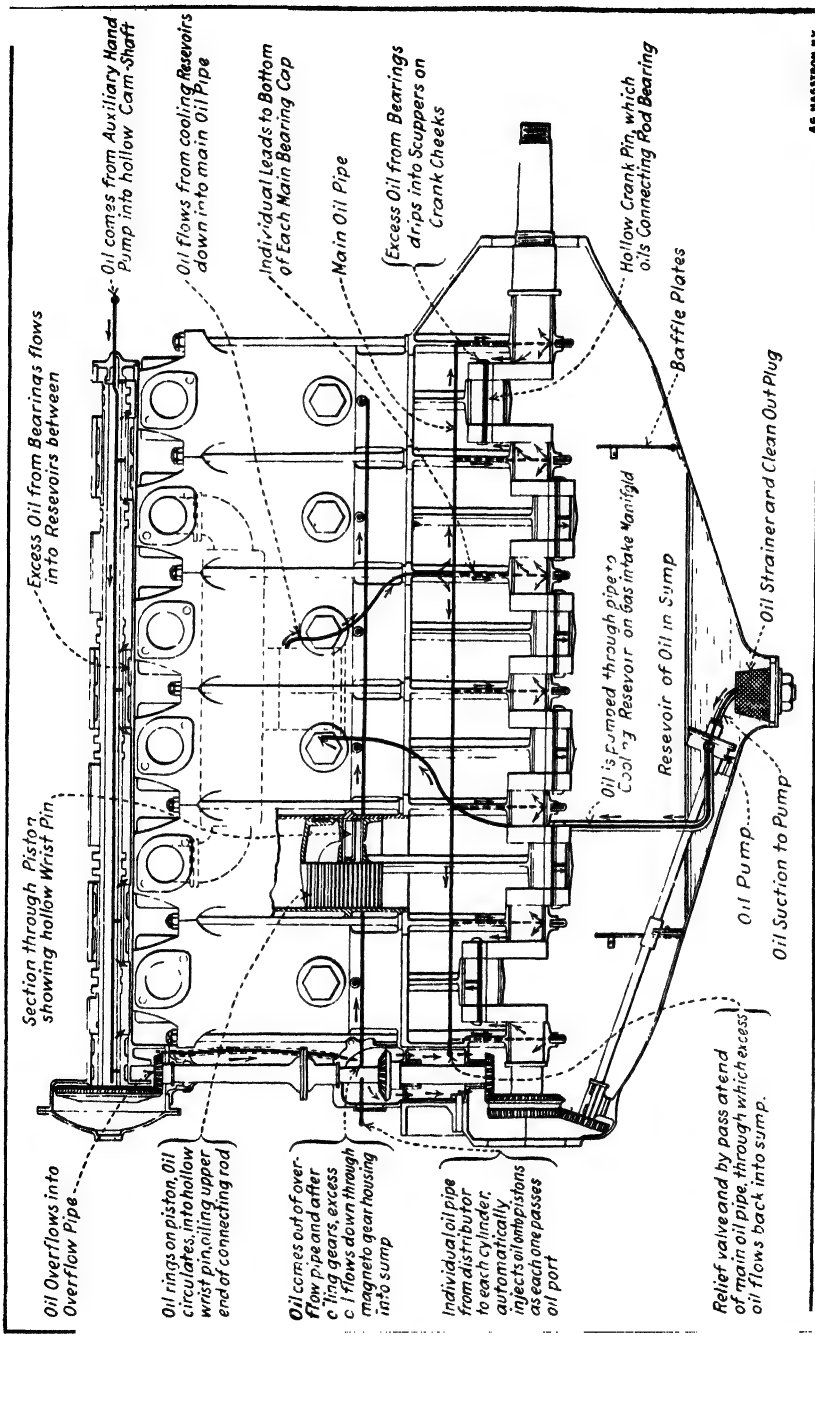


Fig. 209.—Diagram of Oiling System Employed on Early Hall-Scott Type A—Six-Cylinder, 125 Horsepower Engine.

is common practice, but the normal oil level is below the point where it will be reached by the connecting rod. It is drawn from the crankcase by a plunger pump which directs it to a manifold leading directly to conductors which supply the main journals. After the oil has been used on these points it drains back into the bottom of the crankcase. An excess is provided which is supplied to the connecting rod ends by passages drilled into the webs of the crankshaft and part way into the crankpins as shown by

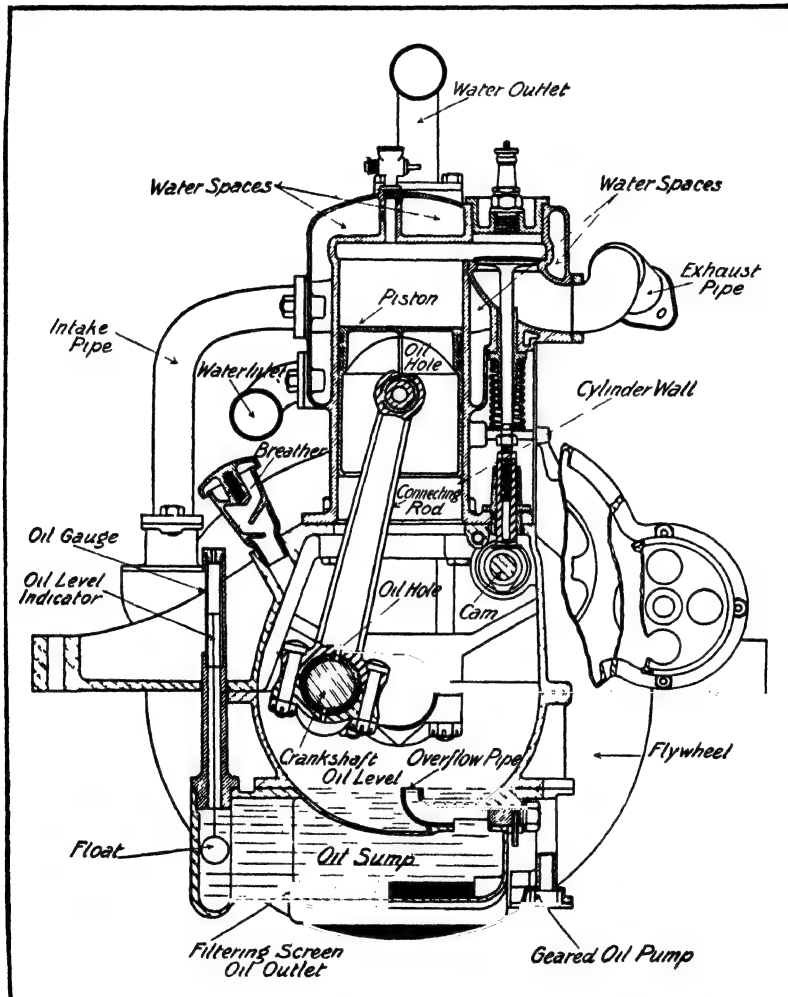


Fig. 210.—Sectional View of Typical Automobile Motor, Showing Parts Needing Lubrication and the Method of Applying Oil by Constant Level Splash.

the dotted lines. The oil which is present at the connecting rod crankpins is thrown off by centrifugal force and lubricates the cylinder walls and other internal parts. Regulating screws are provided so that the amount of oil supplied the different points may be regulated at will. A relief check

valve is installed to take care of excess lubricant and to allow any oil that does not pass back into the pipe line to overflow or bypass into the main container.

A simple system of this nature is shown graphically in a phantom view of the crankcase at Fig. 211, in which the oil passages are made specially prominent. The oil is taken from a reservoir at the bottom of the engine base by the usual form of gear oil pump and is supplied to a main feed manifold which extends the length of the crankcase. Individual conductors lead to the five main bearings, which in turn supply the crankpins by

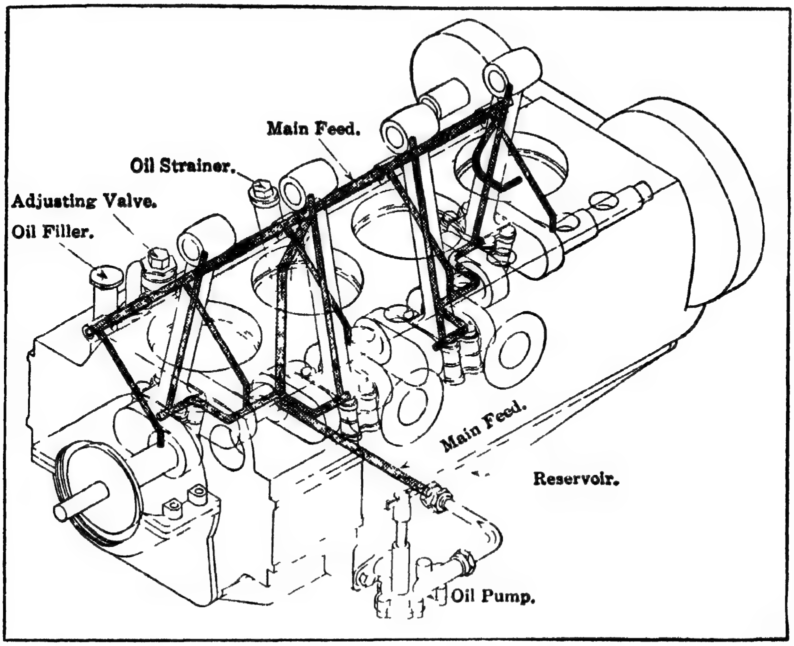


Fig. 211.—Simple Pressure Feed Oil Supply System for Automotive Powerplants Using Only One Pump and Carrying Oil in the Sump.

passages drilled through the crankshaft web. In this powerplant the connecting rods are hollow section built up members and the passage through the center of the connecting rod serves to convey the lubricant from the crankpins to the wristpins. The cylinder walls are oiled by the spray of lubricant thrown off the revolving crankshaft by centrifugal force.

Oil projection by the dippers on the connecting rod ends from constant level troughs is unequal upon the cylinder walls of the two-cylinder blocks of an eight- or twelve-cylinder Vee-engine. This gives rise, on one side of the engine, to under-lubrication, and, on the other side, to over-lubrication, as shown at Fig. 212 A. This applies to all modifications of splash lubricating systems. When a force-feed lubricating system is used, the oil, escaping past the cheeks of both ends of the crankpin bearings, is thrown off at a tangent to the crankpin circle in all directions, supplying the cylinders on both sides with an equal quantity of oil, as at Fig. 212 B.

Oiling Curtiss OX Engines.—The lubrication system used on Curtiss OX type engines is of the wet sump type. The oil is circulated to the main journal bearings, the connecting rod crankpin bearings and the camshaft bearings under a pressure of from 45 to 60 pounds. The cylinder walls, piston pins, thrust bearing and minor enclosed parts are lubricated by spray; the surplus oil draining to the sump in the lower half of the crank-

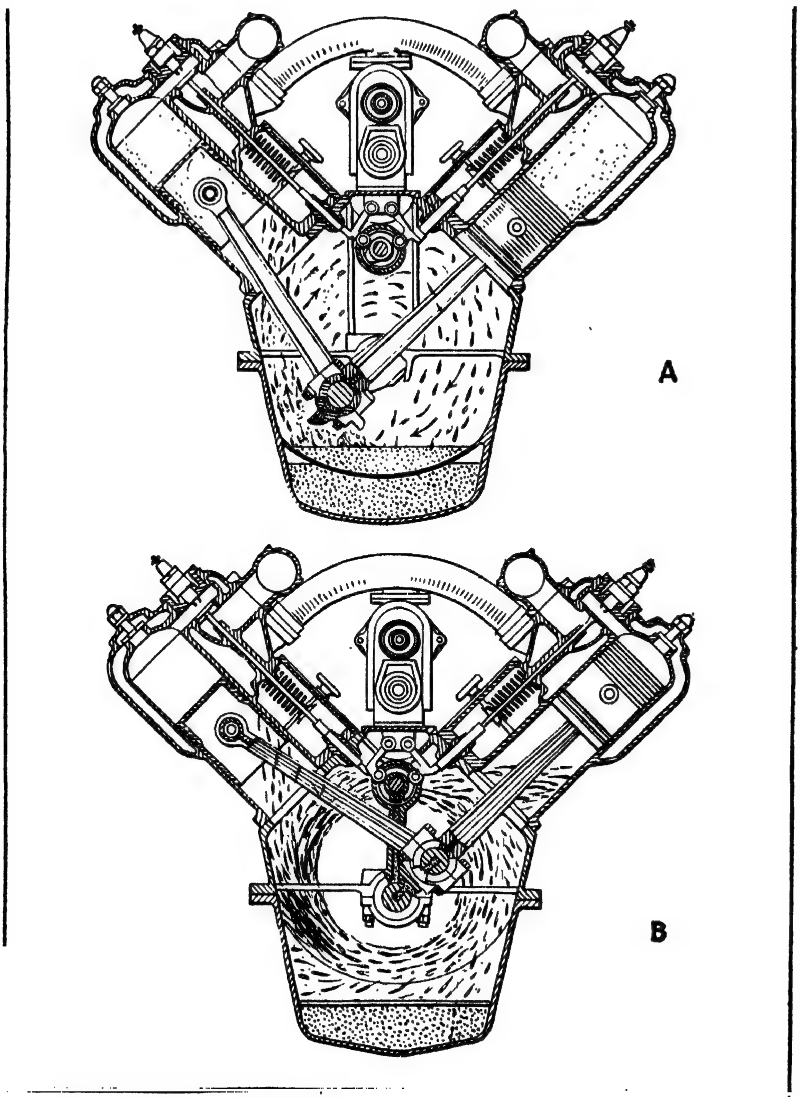


Fig. 212.—Why Dry Sump Pressure Feed Lubrication is Best for Vee Type Airplane Engines. A—When Splash is Used, One Bank of Cylinders Gets More Oil than the Other. B—Equal Lubrication in Both Cylinder Banks Assured by Centrifugal Distribution of Oil.

case where the oil supply is maintained. A gauge on the side of the crankcase indicates the level. The bottom of the sump slopes from front and rear toward the center at which point it passes through a strainer to the oil pump external suction line. Above the sump are two baffle plates, one on each end, which are inclined parallel to the sump bottom and drain into the removable strainer. These baffles prevent over-oiling when flying at steep angles.

The oil pump which is removable is located in the bottom half of the crankcase at the propeller end, and consists of two gears. It delivers the oil to the front end of the hollow camshaft through an external delivery pipe. At this point are located the pressure relief valve and the pressure gauge connection. Return oil from the relief valve is allowed to drip over the crankshaft and oil pump drive gears on its way to the sump. The adjustment of the pressure relief valve is accomplished by varying the thickness of the fiber washer under the head of the screw which bears against the pressure regulating spring. From the camshaft the oil is delivered to annular grooves in the camshaft bearings through holes drilled in the journals. From this point it flows down through holes in the bottoms of the annular grooves, in register with leads cast in the webs of the crankcase, to the main journal bearings. The journals are bored as are the crankpins, a connection between them being made by drilling the crankcheeks. The connecting rod big ends receive their lubrication from the crankpins through holes drilled in the latter. A small quantity of oil is allowed to escape from the rear end of the camshaft, which lubricates the timing gears and magneto drive gear. The piston pin bosses have holes in their upper sides to lead oil to the piston pins which rotate in the bosses.

Oil Pumping and Carbon Deposit.—"Oil Pumping," in the common use of the term, refers to an accumulation of oil in the combustion-chambers rather than to the quantity which actually passes the pistons. With adequate cylinder lubrication, there is normally a certain quantity of oil passing into the combustion-chambers. If it is burned, its presence is not objectionable—but if it accumulates, fouled sparkplugs, sticky valves and excessive carbon deposits are likely to result. An engine operating under a heavy load will burn cleanly an excess of oil, while one which is lightly loaded or running idle cannot consume large quantities, particularly if the lubricant is richer than the operating temperatures call for. The result is oil pumping troubles which are always aggravated when an oil heavier than recommended is used.

Wear of cylinders and pistons with increased normal clearances, or wear of the piston rings may be responsible for an excess of oil in the combustion-chambers. Wear of the rings in their grooves will cause a definite pumping action—lifting the oil mechanically into the combustion-chambers. When wear occurs, it must be remedied by renewing or refitting the parts affected. With correct lubrication, wear will be greatly reduced. Carbon accumulation in the engine is the result of incomplete combustion—either of the oil or of the fuel or both. This failure to burn the oil and fuel completely may be due to the lack of sufficient air for complete combustion or to the lack of sufficient heat for proper vaporization.

Oil pumping and excessive carbon deposits in modern engines may be controlled by careful observation of the following suggestions:—

1. See that scavenger pumps are functioning properly.
2. Use a high quality oil of the body and character recommended by the engine manufacturers. Either an incorrect grade or a poor quality oil is certain to make trouble.
3. Do not try to compensate for wear by using a heavier bodied oil than is recommended. The heavy oil when heated will pass the pistons almost as readily and will be harder to burn. The trouble will therefore be aggravated instead of corrected.
4. If the oil pressure falls off gradually, a possible cause is worn bearings, which allow too much oil to be sprayed from the bearing clearances. If this is the case, it is obviously wrong to try to correct the condition by increasing the pressure and feeding still more oil, or by changing to oil of a heavier grade. Oil diluted by fuel will cause a falling off in the oil pressure. It is therefore advisable to drain the crankcase completely and refill with fresh oil before concluding that the bearings are at fault.
5. Be sure that the carburetor is not feeding too rich a mixture. If there is not enough air to consume all the fuel, there certainly will not be enough to consume any excess oil which passes into the combustion-chamber. Incomplete combustion means carbon.
6. "Missing" promotes oil pumping and carbon formation because the oil normally passing into the combustion-chamber is not burned. Keep the ignition system in good condition.
7. Compression losses affect the efficiency of the engine and the complete combustion of oil and fuel. Keep the valves properly ground in and the tappet clearance properly adjusted.

Sludge.—Sludge, though not particularly prevalent in aircraft engine operation, may become troublesome under short flights or extremely low temperature conditions to the extent of clogging some of the smaller oil passages or the oil screen. Sludge is an emulsion of oil, water and impurities which accumulate in an engine run too cold. Water vapor constitutes a large percentage of the exhaust gas in normal combustion. As long as the cylinder wall temperature is above 120 degrees Fahrenheit this vapor passes out the exhaust port and does no harm, but with a comparatively cold cylinder wall this vapor condenses on the oil film and some of this moisture is scraped into the crankcase by the piston rings on each downward stroke.

Agitation of the water, oil and impurities by the circulating pumps whips these substances into a permanent mixture or emulsion. When this emulsion is forced into the whirling crankshaft centrifugal force separates the sediment from the mixture and deposits it on the outer walls of the drilled passage. If this deposit becomes excessive, it may clog the lead to the master rod bearing or the leads to the cam disc bearing and the magneto drive shaft, and cause serious damage. Sediment can be minimized by following the draining instructions of the various engine builders for without impurities emulsion and sludge are impossible.

Rust-Corrosion.—Occasionally some of the polished parts of engines, such as piston pins and valve stems, are found to be rusted or corroded. This trouble is due, first, to the presence of water in the crankcase and, second, to the fact that badly diluted oil does not protect the working parts from the rusting action of the moisture. If this moisture is made acid, as it can be through the use of fuels containing excessive amounts of sulphur, the surfaces may become corroded very rapidly. Any sulphur which is contained in the fuel burns in the cylinders and forms sulphur trioxide (SO_3). If there is leakage past the pistons and rings, part of this sulphur trioxide will find its way into the crankcase along with a considerable quantity of water vapor,—one of the products of combustion. When the engine is cold, this water vapor will condense into liquid form and unite with the sulphur trioxide to form sulphuric acid (H_2SO_4). If the crankcase oil is badly diluted, it will drain off of the parts, leaving them exposed to the action of this acidulated moisture which of course, tends to corrode them. Even if the fuel is free from sulphur compounds which would form acid, the parts may rust due to their becoming coated with moisture.

Rusting and corrosion troubles may be avoided by observing the following precautions, as there can be no rusting or corrosive action if the parts are protected by oil:

1. Minimize dilution by using the mixture control to keep the fuel consumption at its most economical value.
2. Keep the engine in such mechanical condition that the burning gases will not readily pass the pistons and rings.
3. Use the correct oil and keep it in good condition so that the pistons will be sealed against leakage.
4. Follow the draining suggestions given in this book and in the instruction manuals of the various engine builders and oil companies.

QUESTIONS FOR REVIEW

1. Why is lubrication necessary in bearings?
2. What is the theory of lubrication?
3. What are the main requirements of oils?
4. Where do lubricants come from?
5. What vegetable oil is used for engine lubrication and to what type of engine is it best adapted?
6. What are mixed oils; mineral oils; organic oils?
7. Describe simplest method of engine oiling.
8. What is the best oiling system for modern aircraft engines?
9. Name parts and operation of dry sump oiling system.
10. What causes sludge; rust, corrosion?

CHAPTER XVII

MODERN AVIATION ENGINE LUBRICATION SYSTEMS

Aviation Engines Present Difficult Problem—Faults of Force Feed Oil Systems—Effect of Varying Clearance—Oiling System of Wasp Engines—Whirlwind System of Lubrication—Hispano-Suiza Oiling System—Liberty "12" Oiling System—Maybach Engine Lubrication—Isotta-Fraschini V6 Oiling System—Farman Inverted Engine—Lubrication of Anzani Engines—Efficiency of Oil Pumps—Fresh Oil Systems—Temperature Effect on Power Delivery—Hispano-Suiza Oil Cooling System—Oil Temperature Control Not Only Solution—Wright Oil Temperature Control System—Packard Oil Radiator—Oil Cooling by Intake Gas—Ball and Roller Bearings Have Little Friction.

The selection of the correct grade of oil for an engine depends upon the consideration of every feature of its design, construction and operation, which may affect either the ability of the oil to maintain an adequate lubrication film on all working parts, or the performance of the oil in the engine. Certain factors such as heat and dilution by unvaporized fuel tend to thin out the oil, reducing its protective effect, while the pressure of the gases above the pistons tends to force the oil from the clearance spaces, thus impairing the piston seal. Under some conditions, these factors may be of minor importance; under others, they may have a marked effect on the lubricating oil. In the first case, light bodied clean burning oils of high quality will assure adequate lubrication; in the second, a high quality oil of heavy body and rich lubricating value, may be very desirable in order to offset the effect of these factors to the maximum degree.

In contrast to these considerations, there are others which, at times, may prevent the use of heavy bodied, rich lubricating oils. For example, to assure reliable distribution of the oil by the lubricating system; to minimize power loss and fuel waste due to the internal friction of the oil itself; or to reduce carbon deposits to a point where they are detrimental to satisfactory engine performance, in some instances, it may be necessary to use light bodied oils of such character that they will be distributed readily and burned cleanly under the most unfavorable conditions. The relative importance of these factors varies widely in individual engines, and many apparently minor features of design, construction or operation often have an important bearing on the lubrication requirements of a particular unit.

Aircraft Engines Present Difficult Problem.—Few, if any other, machines or engines present as difficult a lubricating problem as is found in the lubrication of aircraft engines. In the solution of this problem, consideration must be given not only to the type of engine and the method of lubrication employed, but also to the wide range of conditions under which it must operate, all of which affect lubrication. Aircraft engines are expected to operate perfectly under full load in hot summer weather at sea level atmospheric pressure, on the one hand, and at idling speed in subzero temperatures at an altitude of 10,000 feet or more, on the other. It has not been possible to design temperature controlling devices, either coolers or radiators, in the lubricating systems which will compensate

entirely for such a wide range of conditions and the lubricating oil must, therefore, be able to withstand wide and rapid variations in temperature, maintaining during the entire time an almost frictionless film of oil on the surfaces of the moving parts. Local conditions also have a pronounced effect on the oil film so necessary in aviation engines. The steel cylinders used in a number of engines are very thin and are exceptionally good conductors of heat. There is always a possibility of the part of the water-cooled cylinder-walls near the water inlet remaining relatively cool, while the area near the exhaust may reach extremely high temperature.

The pressure gauge is usually connected to the main oil lead from the pump and the pressure registered on it, shows only the resistance, in pounds per square inch, to the flow of oil through the bearings. It is conceivable that through some stoppage of the oil lead, the pressure might be greatly increased even though less than the normal amount of oil was actually reaching the bearings. It is also evident that in the case of an old or worn motor, bearing clearances will have increased until they offer very little resistance to the flow of oil and, as a consequence, the pressure will be low, yet it is under just this condition that the bearings are receiving the greatest amount of oil. Unless all of the other factors affecting lubrication are known to the pilot, the pressure gauge is not a safe index as to the lubrication of the bearings of his engine.

Heating an engine before starting in cold weather, if only by filling the circulating system with hot water, is a matter of prime importance. It is equally as important to drain the lubricating system at the end of each flight and to refill it with warm oil before starting again. In cold weather, an airplane engine after starting should invariably be run on closed throttle until thoroughly warm, before taking off. The fact that tardy action of the oiling system is often due to failure of the heavy oil to flow through the pump screen has not been often mentioned. A conventional gear-pump will draw all the oil from inside a pump screen in a few seconds after starting and, if the oil does not flow through the screen rapidly enough to supply the pump, the pump will suck air when any portion of the screen is near the oil-level. The oil-level in this sense may mean an irregular outline representing the top surface of the semi-fluid oil, usually cupped-out in the vicinity of the pump immediately after starting in cold weather. To prevent the pump from drawing in air and to force it to exert a suction on the oil surrounding the screen, an air-tight bell is sometimes fitted over the screen, so that air cannot flow through any portion of the screen that otherwise would be exposed above or near the oil-level under the conditions already described.

Faults of Force Feed Oil Systems.—The conventional force-feed system has one other weakness in that, on a new engine, with the bearings fairly tight, a minimum amount of oil is thrown-off to the cylinders, which need oil at this time more than at any time during the life of the engine. Later on, cylinders and pistons acquire a good polished finish and less oil is required for their lubrication than when they left the factory. The greater clearance in the bearings due to wear permits more oil to escape at this time, when really less oil would be desirable because worn or "run-in" bearings run with less heat and friction than tight ones. In

extreme cases this permits bad cases of over-lubrication, the effects of which are exaggerated still further due to the decreasing efficiency of the average piston ring. By this is meant both the wearing of the ring in the piston-ring groove and the tendency of the face of the ring to round over. A ring worn thus has obviously lost its most valuable asset, that is, its ability to scrape off surplus-oil on the down stroke, and an examination of rings from engines inclined to pumping oil will often show this wear at the edges of the face, which are no longer sharp as on a new ring but somewhat chamfered.

The quantity of oil thrown off by the crankshaft cheeks can be controlled fairly well by proper main-bearing construction, in which the oil-groove in the face of the bearing does not extend into the shim contact-surface. There is a mating groove on the back of the bearing, the groove in the upper and lower halves registering through a hole in each shim. The inner groove on the upper half of the bearing is thus supplied with oil under pressure, but no oil is under pressure at the points where the shims usually seat against the crankshaft, the point where most leakage takes place.

Lubrication of pistons and cylinders on high-speed engines is further complicated by a condition that arises from the nature of the finish of the cylinder-walls. If the walls are not absolutely smooth but contain minute corrugations or other irregularities due to the method of finishing, the rings will be forced away from the cylinder-walls at high piston-speeds. This tendency of rings to collapse permits the hot gases to blow down past the piston-walls, and is detrimental to the lubrication of the piston, cylinder and rings.

The Effect of Varying Clearance.—The effect of varying clearance between the moving parts of an airplane engine on lubrication is stated thus by the Equipment Division of the U. S. Air Corps:

"The clearances that are allowed for bearings lubricated by means of a force feed system must be considered in the light of a check or throttle valve at the outlet of the system. Close clearances have the effect of checking the flow of the oil by reducing the outlet; wide clearances allow a free passage of the lubricant. Close clearances are always accompanied by high pump pressures, as compared with wide clearances with the same lubricant. Therefore, with the same lubricant, higher pump pressure can be secured by reducing the clearances. With the reduction of clearance, there is a reduction in the amount of oil that flows through the bearing. With the same clearance it is necessary to decrease the body of the lubricant to increase the flow. In starting an engine with cold oil the pump pressure is high. This pressure falls as the body of the oil is reduced; first by the heat taken from the pistons in the cylinders; second by the heat generated by the oil working through the system and the bearings; and third, by the gasoline that has leaked past the rings and is held by the oil.

"The heat of a bearing not affected by outside temperatures is largely due to the fluid friction of the lubricant, the rule being that the heavier the body of the oil, the higher the heat due to friction. With the same lubricant on a bearing, an increased flow will reduce the heat by simply

carrying it away and not allowing it to accumulate. There is, therefore, an automatic adjustment in the force feed system with the heavy oil—as the bearing heats, the oil feed increases, due to the oil being lighter in body, and this increased feed tends to reduce the temperature of the bearings, and as the temperature is reduced, the body of the oil gets heavier and the flow is checked. It is only necessary to have a large enough supply of oil and a sufficiently large pump at work at any clearance to make overheating almost impossible."

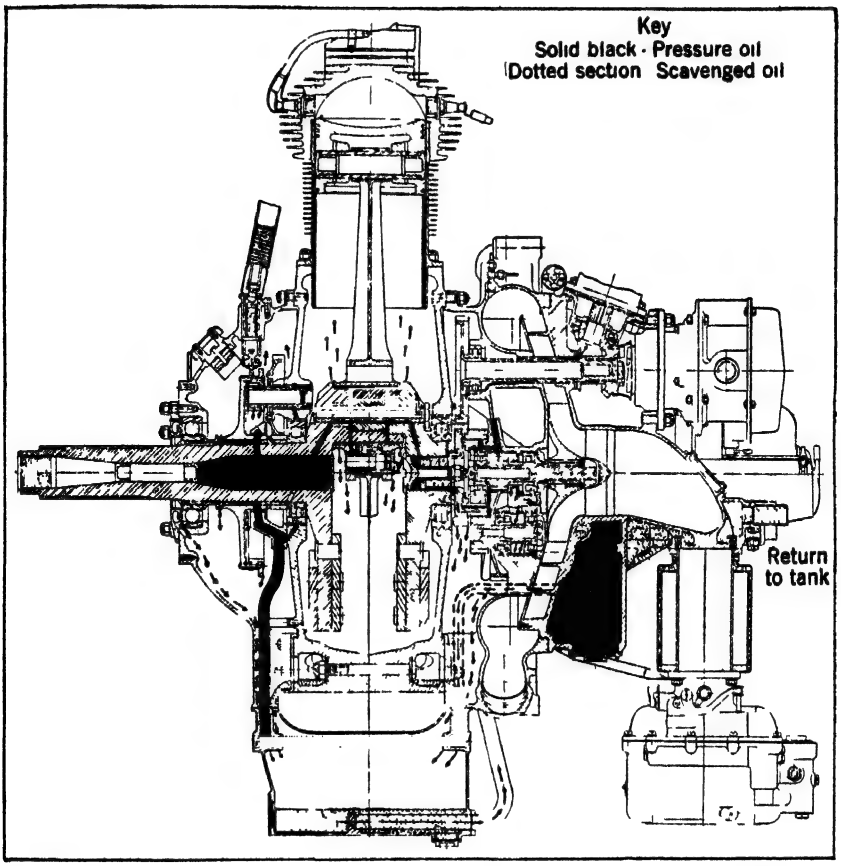


Fig. 213.—Lubrication Chart of the Pratt & Whitney "Wasp" Engine.

Oiling System—Wasp Engines.—The oil tank should be located near the engine, and if possible, slightly above the center of the engine; a large head however, is not desirable since it may cause oil to leak through the pump when standing for a long time and fill the engine. The tank should be protected from the hot air coming from the engine and provision made to pass cool air continuously around it from the slipstream. In designing the oil tank figure on an average oil consumption of one and one-half gallons per hour. This will care for full throttle operation. The filler

should be at least two inches diameter and must be located so as to insure an air space above the oil equal to at least ten per cent of the volume of the oil, as shown at Fig. 215. A three-quarter inch vent should be provided in the top of the tank. If this can be arranged so that oil will not issue from it when the plane is being maneuvered, the vent pipe can be carried down and out of the fuselage. Otherwise, this pipe should be connected to the engine crankcase. A connection for this purpose is provided on the left side of the rear crankcase. A large drain valve should be accessibly located under the oil tank. The oil piping to and from the engine should be one inch seamless copper tubing, and should have as few bends and connections as possible. Smaller piping will give trouble in cold weather when the oil is thick. See the oil connection diagrams for the proper hook-up, these being given at Figs. 213 and 214.

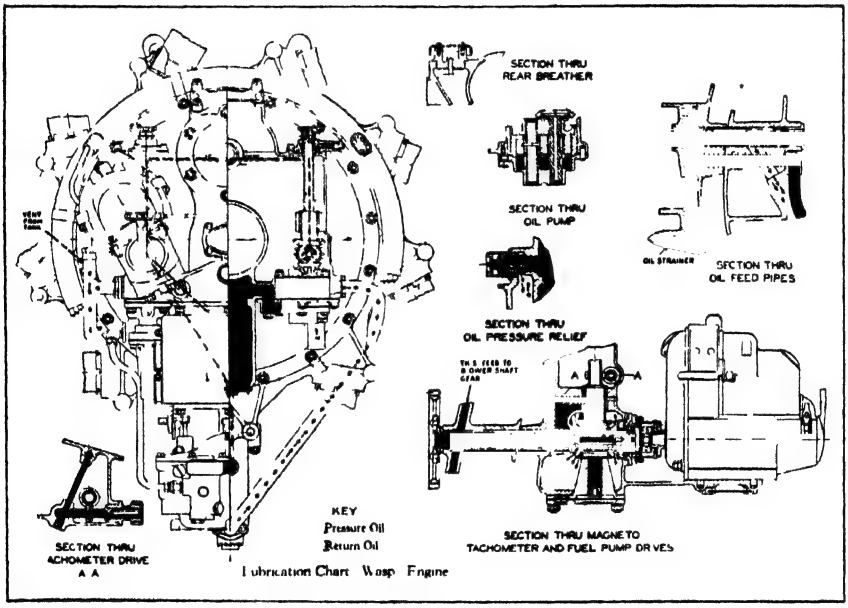


Fig. 214.—View Showing Methods of Lubricating Accessories of the Pratt & Whitney "Wasp" Engine.

The oil strainer is at the rear of the engine just forward of the carburetor. Provision should be made in the cowling and structure to get at this readily. The oil pressure relief valve is at the side of the strainer chamber, and faces toward the rear and the left side and should be made accessible. The oil pressure gauge connection is on the right side of the engine at the rear of the mounting flange and is tapped $\frac{1}{8}$ inch pipe thread. $\frac{5}{16}$ inch O. D. copper tubing should be used with a pig tail at the engine, and a 150 pound gauge. The oil thermometer connection is also on the right side and is in the oil outlet passage. This is tapped $\frac{5}{8}$ inch eighteen threads. The thermometer should read in degrees Fahrenheit to 212 degrees, or degrees Centigrade to 100 degrees.

Whirlwind System of Lubrication.—The force feed system of lubrication is employed in Wright Whirlwind engines, the oil being delivered under pressure to all friction surfaces except the cylinder walls, piston-pins, and accessory drive gears. This system is clearly shown in sectional views of the engine given at Fig. 208, the solid section showing fresh oil from the tank. Unlike the conventional automotive type of engine, the oil supply is carried in a tank separate from the engine. To circulate this oil three pumps of the gear type are employed. These pumps, which are

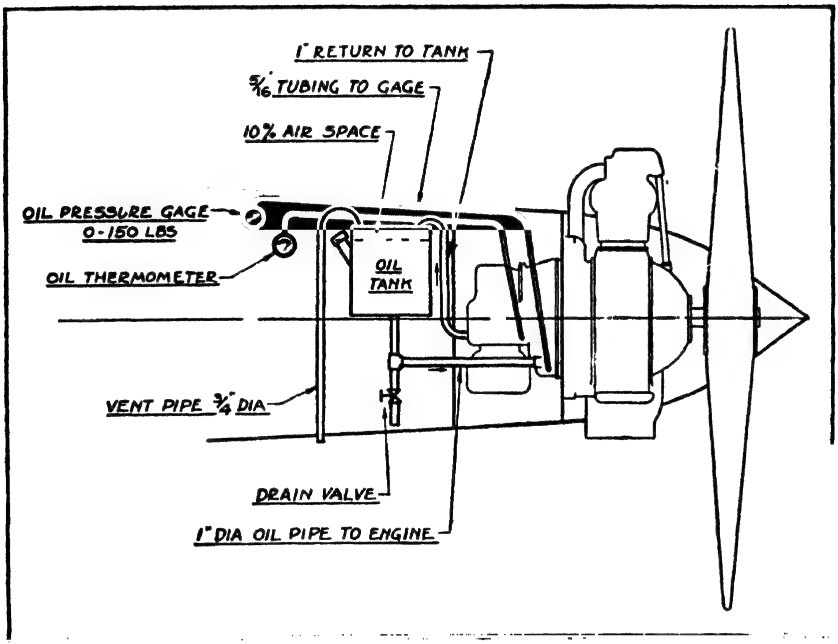


Fig. 215.—Diagram of External Oiling System for "Wasp" Engine.

assembled as a unit and driven by a single shaft, can readily be removed for inspection or cleaning. One of the pumps draws fresh oil from the tank and forces it through a double screen to a circumferential groove in a plain bearing positioned on the rear end of the crankshaft. The oil enters the hollow crankshaft through a hole which registers with the circumferential groove, filling the crankshaft throughout its length. A hole drilled radially in the crank-throw leads the oil under pressure to the master rod bearing from where it is forced through channels in the rear of the bearing and drillings in the flanges of the bearing shell to the link rod knuckle pins.

Other openings drilled in the hollow crankshaft lead the oil under pressure to the cam disc bearing, the hollow cam drive shaft and the magneto drive shafts. The oil sprayed off of the master rod bearing and the link rod knuckle pins is thrown to the cylinder-walls and piston-pin bearings providing ample lubrication for these parts. The excess oil from the cam and magneto drive gears and drive shafts falls to the bottom of the front section and drains off through an external pipe to the sump. Surplus

oil in the crankcase main section also drains into the sump but through a passage cast in the crankcase and in the intake manifold. The sump is located in the inlet manifold and consists of a jacket surrounding the three induction passages. It serves not only as a drainage point for the oil but also as a heater for the incoming fuel mixture. Oil collecting in the sump is drawn up by one of the scavenging pumps through an external line and delivered to the oil return line. The oil spraying off the rear crankshaft bearing, the oil from the accessories drives and the discharge from the pressure relief collects in the bottom of the crankcase rear section. Oil at this point is drawn off by the second scavenging pump and is also delivered to the oil return line.

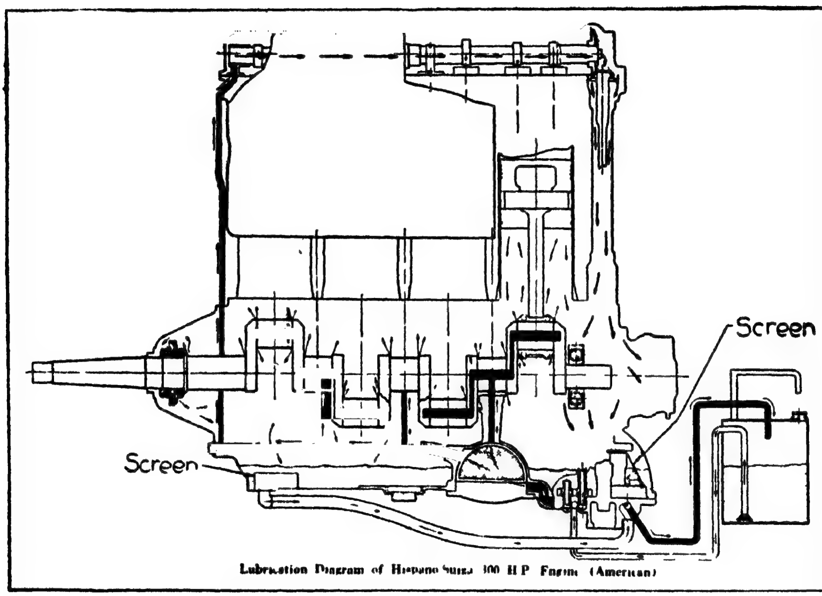


Fig. 216.—Lubrication Diagram of the Hispano-Suiza 300 Horsepower Engine.

Hispano-Suiza Oiling System.—American Hispano-Suiza 300 horsepower Oiling System: Lubrication of this engine is by the dry sump, force feed type in which oil is supplied to main bearings under pressure and to cylinder-walls and wristpin bearings by oil spray from the rotating crankshaft. The oil pumps are located in the rear of the lower crankcase. There are two scavenging pumps and one pressure pump, all of the gear type. The simplified diagram from the *Air Service Engine Handbook* at Fig. 216 shows the system very clearly. Oil is drawn from an outside tank by the pressure pump and forced under pressure through an oil screen which is enclosed in a hemispherical aluminum housing, integral with the lower case and situated directly under crankshaft bearing number four. After passing the screen the oil is conducted by a steel tube, cast in the crankcase, to the three front main bearings. The fourth bearing has a direct lead from the oil filter compartment. Oil passes from a circumferential

groove in back of each main bearing to the bearing surface through four equally spaced holes. The crankshaft is hollow with radial holes in the journals and crankpins, providing lubrication for the connecting rod big end bearings. Holes are provided in the marine rod bearing so that the oil is properly distributed to the outer or plain rod bearing. Excess oil thrown off from the main and connecting rod bearings lubricates the piston-pin bearings, pistons and cylinder-walls.

Through a bypass around the propeller end bearing, oil is led to tubes running up the front of each cylinder block to the front camshaft bearing. A hole in the bearing in register with a hole in the journal, allows the shaft to fill with oil. Oil is led to the remaining camshaft bearings through drilled holes in the camshaft. Small holes in the cams supply oil to the cam follower faces. Oil from the camshaft housing flows down the inclined drive shaft housings over the driving gears and rear crankshaft ball-bearing, the excess oil gravitating to the rear of the lower case. The bottom of the lower case acts as an oil sump, from which the oil is removed as rapidly as it collects. One of the suction pumps draws the oil from the front of the sump, the other pump draws it from the rear and both the pumps deliver into a common discharge pipe connected with the oil tank. An oil relief valve is accessibly located on the side of the crankcase from which it extends into the strainer chamber. When the pressure exceeds an amount regulated by the spring tension, the valve is lifted from its seat allowing oil to bypass from the strainer chamber directly into the crankcase. There is no adjustment on the spring tension but the spring is made of such a length that the valve will open between 60 and 85 pounds pressure.

Liberty "12" Oiling System.—The lubricating system is of the dry sump, pressure type. Oil is supplied under pressure to the main bearings, connecting rods, and camshafts. The oil pumps are located at the rear of the engine and are assembled with the oil strainers in an aluminum housing which bolts to the under side of the lower crankcase. There are one pressure pump and two scavenging pumps of the gear type as shown at Fig. 217. The oil is drawn from an outside tank or radiator by the pressure pump, through a large filtering screen and delivered into a steel manifold running the whole length of the lower crankcase. Leads are carried from this to each crankshaft bearing and thence through passages in the crankshaft to the crankpins. The leads to the main bearings terminate in annular grooves in the bearing linings, which extend approximately one-sixth around the circumference. Consequently, during the time that these grooves are in register with the holes in the journals, the crankpins are under pump pressure. During the remaining five-sixths of the time they are dependent for their oil supply entirely upon the pressure created by centrifugal force. The cylinders, pistons and piston pins are lubricated by the oil swirl of the crankshaft. A passage is provided around the propeller end bearing from which two oil leads are taken to the camshafts. The oil is carried in at the front ends of the camshafts, passes through the camshaft bores and is distributed to each camshaft bearing by properly placed holes. The rocker arms are lubricated by surplus oil from the bearings. All the excess oil in the camshaft housing eventually returns to the sump

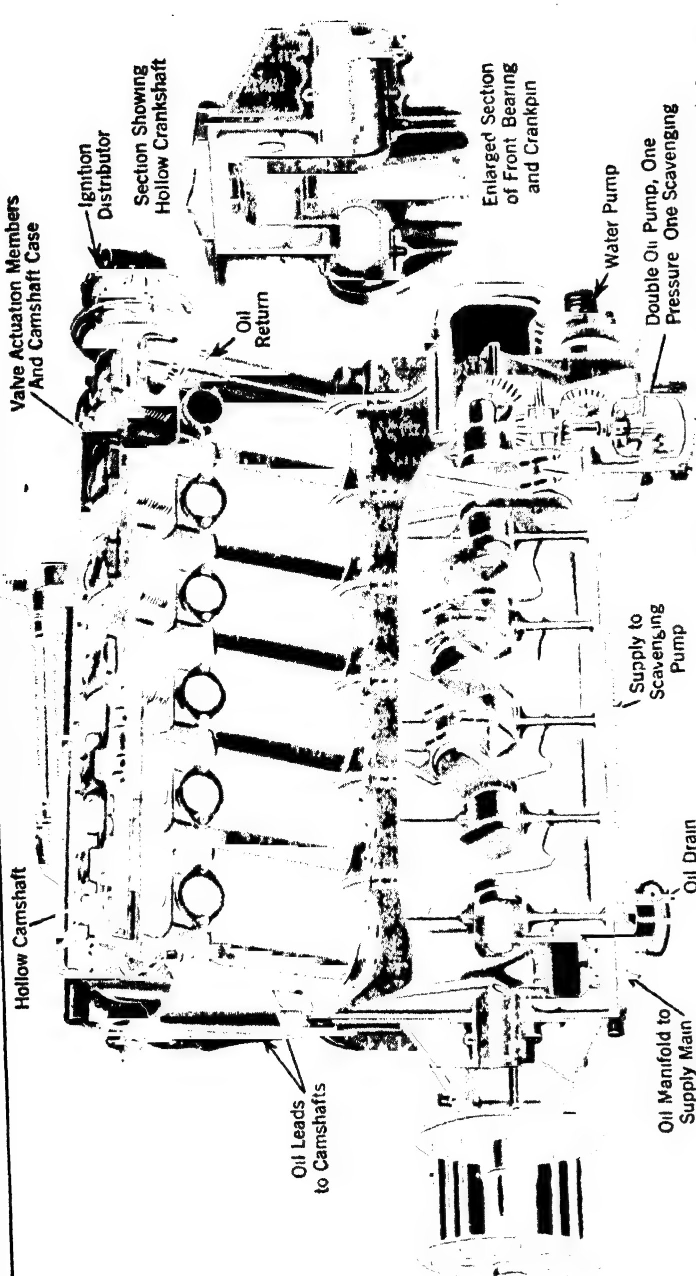


Fig. 217.—Oiling System of the "Liberty" Aviation Engine Operated on Dry Sump Principle, Oil from the Pressure Pump Going to Bearings and Camshaft, the Oil Spray Thrown Off by the Crankshaft Lubricating Interior Walls of the Cylinder and Other Parts. Suction or Scavenging Pump Draws All Oil from the Crankcase.

through the housing of the camshaft drive, oiling the gearing on its way. One of the scavenging pumps draws from the front and one from the rear of the lower crankcase. They return the oil to the supply tank through a common lead. A relief valve is provided between the pressure pump and the oil manifold which bypasses the oil back to the suction line of the pump, when the pressure exceeds the desired amount. This relief valve has a fixed setting. The tension of the relief valve spring can be adjusted by the insertion of shims. The valve itself has four small holes drilled through

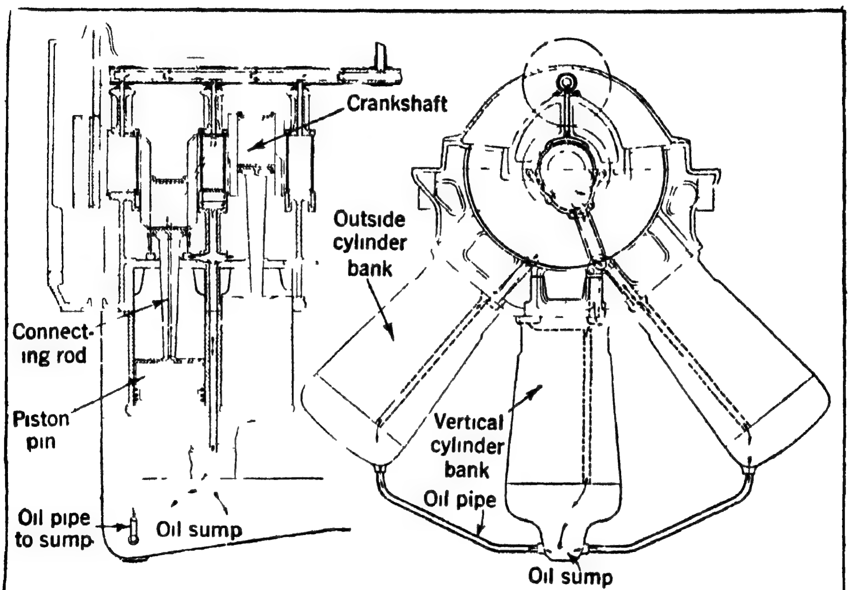


Fig. 218A.—Oiling Diagram of the Farman Eighteen-Cylinder Inverted "W" Type Motor.

it at the stem which bypass a small, constant quantity of oil to prevent over-lubrication at idling speeds. The thrust bearing and tachometer drive are each provided with exterior oil cups for manual lubrication which supplements whatever splash lubrication they may receive.

300 Horsepower Maybach Lubrication.—The lubrication system is of the usual dry sump pressure type. The three separate and similar oil pumps are located in the bottom of the crankcase lower half and driven by a simple horizontal shaft. Each pump consists of a bronze and a steel gear enclosed in a cast-iron case. The front and middle pumps are scavenging pumps and draw oil from two small sumps located respectively at the front and rear of the crankcase bottom. Each sump is surmounted by a small standpipe with a mushroom cover. A slight depression around the standpipe serves as a sediment trap. The delivery ports of the two scavenging pumps are connected by a steel pipe, which runs parallel with the pump shaft on the inside of the case. The outlet from this pipe to the oil tank is on the rear suction well cover. The rear pump, which is the pressure unit, draws its oil supply from an external tank and delivers it past a

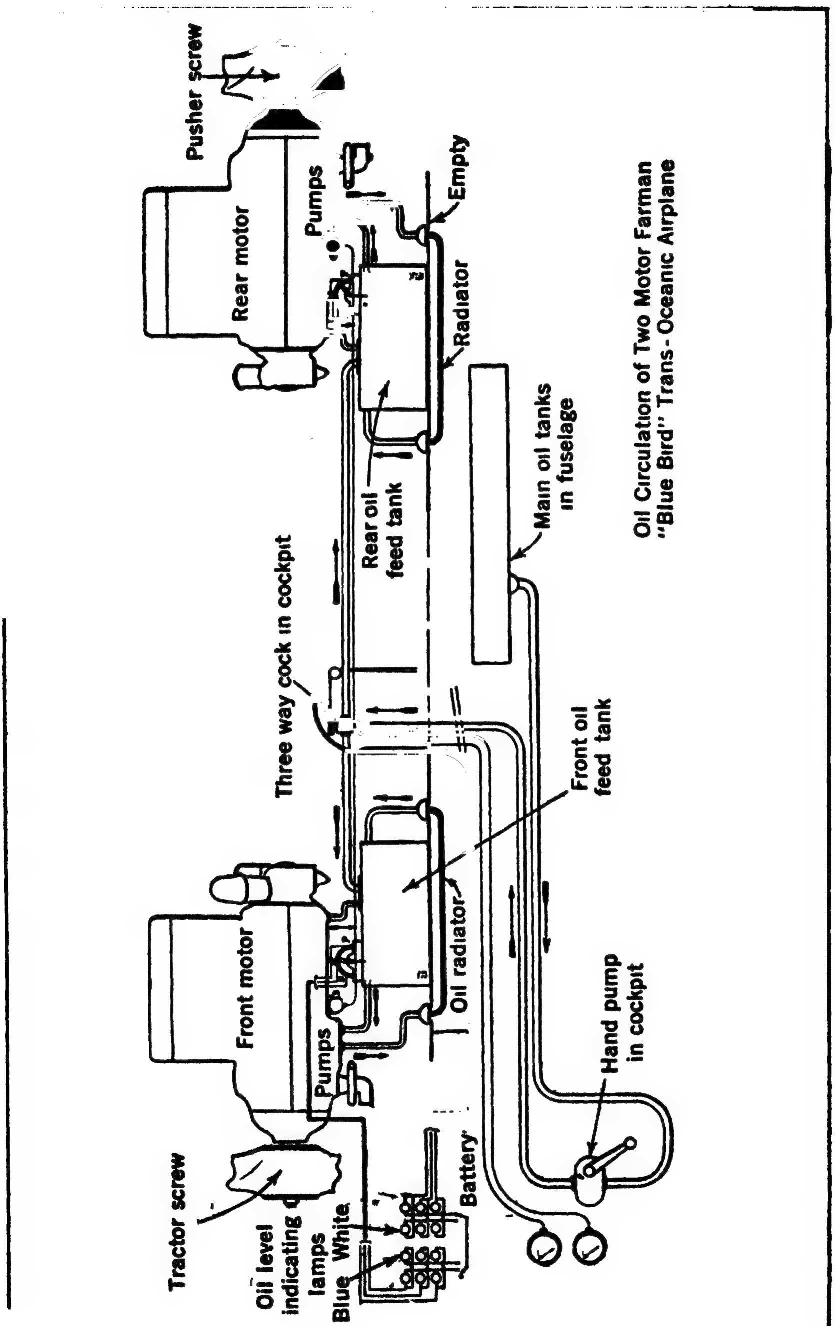


Fig. 218B.—Oiling System of the Farman "Bluebird" Airplane Showing Lines and Also Lamps to Indicate Amount of Oil Available in the Supplementary Oil Feed Tanks.

pressure relief valve and a check valve to the main pressure lead. The relief valve is of the loaded spring type and is adjustable from the outside by means of a thumb screw. It returns surplus oil to the suction side of the pressure pump. The check valve is similar to the relief valve, but has a weaker and nonadjustable spring. Its apparent purpose is to keep the oil passages full while the engine is shut down.

The oil from the pressure pump is distributed to the crankshaft journals by an external oil pipe, located along the side of the upper crankcase. Oil leads are drilled in the main bearing webs of the upper case from the external pipe to the grooved crankshaft bearings. At each connection, between the oil pipe and the crankcase leads, is located an oil metering jet, surrounded by an easily removable filter screen. At the front end of the pressure header, two small leads are run to the camshaft driving gears. These terminate in small jets playing on the gear teeth. The crankpin and the piston bearing are lubricated by centrifugal pressure. This is accomplished by collecting the oil forced out from the main journal bearings in scuppers bolted to the outside of the crank arms. The oil is led by the scuppers into the hollow crankpins, thence through small holes drilled in the crankpins to the connecting rod big end bearings. It then passes up through leads inside of the hollow connecting rods to the piston pin bearings. It is distributed across the perforated piston pin bushing by means of helical grooves inside the connecting rod small end. The lead to the oil pressure gauge is taken from the main oil pipe a few inches back of the last cylinder. Near this point a small pipe is also led to the gasoline pump. The camshafts, cam followers, accessory drive gears and cylinder-walls are splash lubricated by the surplus oil thrown from the crankshaft and oil scuppers.

Isotta-Fraschini V6 Oiling System.—Lubrication is by the pressure dry sump system. A triple vane type pump is built into one assembly, and located at the rear of the engine. The upper unit supplies the pressure feed while the two lower units are the scavenging pumps. A pressure relief valve is attached at the delivery side of the pump which shunts oil back to the sump when a given pressure is exceeded. The oil is drawn from the external supply tank and forced by the pump through an internal main lead to the hollow crankshaft, and an external lead to the hollow camshaft. Holes drilled radially through the camshaft serve the cams, cam followers and bearings. Excess oil flows back through the vertical shaft housing at the rear of the engine, lubricating the driving gear; also through an overflow pipe in the front of the engine. The oil which is transmitted to the crankshaft lubricates the main bearings, and is carried by centrifugal force to the crankpin journals and bearings. A lead through each hollow connecting rod supplies the wristpin. The excess oil flows back into the sump, is drawn off by the scavenging pumps, and returned to the tank. The intake of one of the scavenging pumps is submerged in a small well at the rear; while the other pump receives its supply from the front through an internal suction lead.

Farman Inverted Engine.—The oil circulation, as will be seen from the diagram at Fig. 218 A, is under pressure to the seven main bearings, to the main connecting rods and to the auxiliary rods. By reason of the projec-

tion of the cylinder barrels into the crankcase, the oil from the main and connecting rod bearings is collected around the base of the central group of cylinders, from which chamber it is returned to the tank by the scavenging pump. At each end of the cylinder blocks there is a flow of oil, by gravity, to the aluminum housings enclosing the valve gear. As the central block of cylinders is lower than the two lateral rows, the central camshaft housing forms the general collector into which the lateral blocks drain, and from this point the oil is aspired to the tank. A patented feature of the oil pump allows the area and weight of the pump to be cut in two by making two pinions fulfill three functions; two suctions and one pressure delivery. Thus a single pump housing containing three pinions makes it possible to realize the five following operations: suction front, suction rear, return to tank, suction from tank, delivery to engine.

The oiling system of a Farman airplane, the Blue Bird, which uses two motors for power is shown at Fig. 218 B. A main reservoir of oil is carried in the fuselage from which the lubricant may be supplied to either of the motor feed tanks by a hand pump, the control being by a three way valve. A system of lamps is installed so the level of oil in the feed tanks can always be determined. Blue lamps burning show full tanks, white lamps indicate about half tank while the red light indicates that the oil level is dangerously low.. Two pressure gauges show if oil is circulating through the system, one being installed in the piping of the front motor, the other in the oiling system of the rear motor.

Lubrication of Anzani Engines.—Following a series of experiments at the Anzani factory, the manufacturers of the Anzani engines offer to owners of older Anzanis the following hints whereby they can bring the oiling systems of their engines up to the efficiency of the new engines, and to purchasers of new Anzanis the instructions necessary for the installation of their engine. The lubrication system of Anzani radial air-cooled aircraft engine has two major functions to perform, viz.: to lubricate the motor and to carry off excess internal heat. The usual manner of performing these functions is to pump oil into the crankshaft under pressure, from whence it slings off of the crankpin and bathes the inside of the engine. Part of the oil collects in the bottom of the crankcase and is pumped back to the tank and part is vaporized and beaten into a spray and exhausted from the breather pipes.

In order to gain a maximum of cooling effect with minimum oil loss, many schemes have been evolved to allow the passage of a maximum of oil through the crankshaft with a minimum amount to the interior of the motor. As the least bearing wear upsets all of these schemes and results in excessive oiling and breather loss, Anzani engineers abandoned them after experiment. The original lubricating scheme consisted of a gravity feed oil tank feeding into a chamber by gravity; from the chamber a cam driven plunger pump forced metered quantities of oil to the main bearings and through drilled crankshaft passages to the crankpins, from whence the oil was thrown off and lubricated the interior of the engine. The pump installation and the method of driving it is shown at Fig. 219 A. After performing its lubricating function, it vaporized and blew out of the breathers, thus filling its cooling function. This system was very satisfactory,

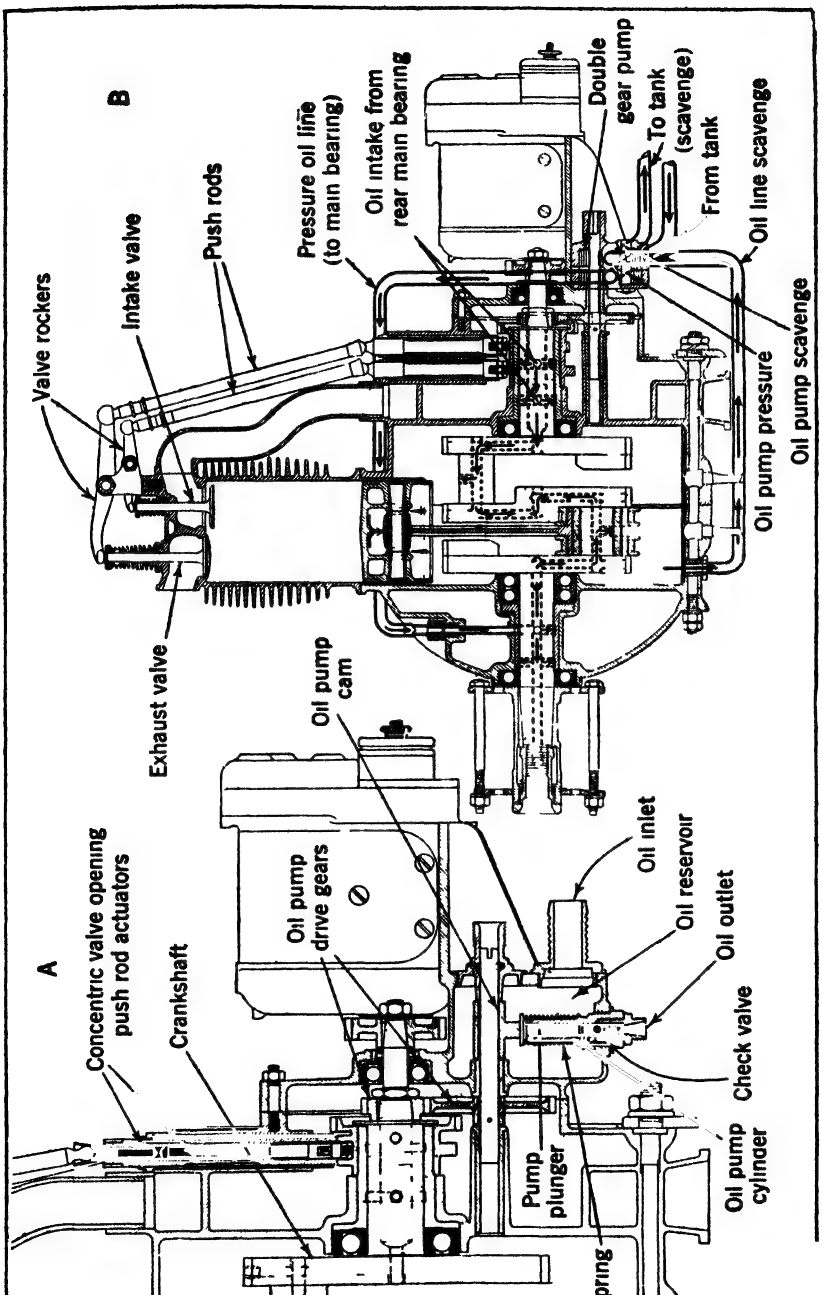


Fig. 219.—Diagram Showing Lubrication of Anzani Aviation Engine. A Shows Plunger Pump Provided on Engines Prior to 1928. Design Shown at B Employs a Double Gear Pump and Operates on the Usual Scavenging System.

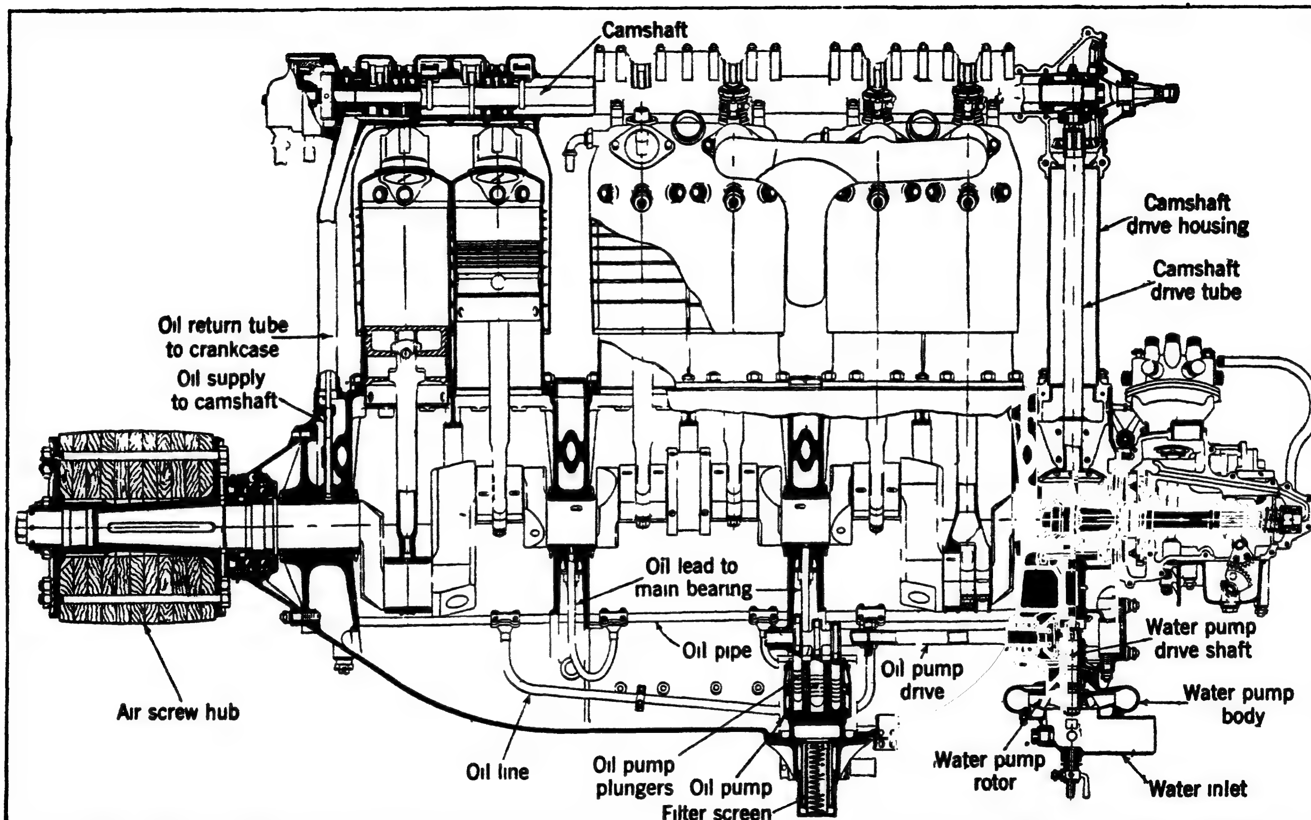


Fig. 220.—Longitudinal Sectional View of Crankcase of the Lorraine Aviation Engine, Showing Location of Triple Plunger Oil Pump, Method of Drive by Eccentrics on Shaft and Also Oil Leads from Pump to Main Bearings. This View Also Shows Location and Method of Drive of the Water Pump.

though wasteful of oil, which caused some operators to evolve various schemes for cutting down the oil supply. This reduced the supply of oil for internal cooling and resulted in overheated pistons and bearings.

After much experiment, the following system was evolved: Remove the breather pipes from the top of the crankcase and plug the holes, remove the drain plugs from the bottom of the crankcase and put a collector tank below this level, connecting it with these drain openings. Fit the breather pipes to the top of this tank on extension pipes, carrying them high up. The excess oil will now drain into this tank and as all breathing is done through it, all oil ordinarily lost as vapor and spray will condense and collect in this tank, from whence it may be pumped to the gravity "nourrice" tank by a hand pump or by a windmill or power driven return pump up to an overflow level in the "nourrice" tank. The engines, up to and including 1927 models, are fitted with the well-known plunger pump, while the 1928 models will be fitted with the new rotary pump with a controlled bypass. With the system described, always operate the oil pump at full feed, viz.: full stroke on plunger pump (no spacer gasket under body flange) and with the bypass handle of the new pump pointing towards the letter "O" stamped on the cover. When the plunger pump is removed for any cause whatever, be sure to inspect the little check valve at the base and see that the oil aspiration holes in the base of the pump are clear. It is also a good plan to stretch the spring to insure the plunger following the operating cam. With the ordinary oil a pressure gauge placed in the line between the pump and the oil feed into the bearings should give a reading of five to six pounds, which will drop somewhat with a hot engine. At the Anzani factory the engines are tested with the highest obtainable grade of genuine cold pressed Castor oil, as this lubricant is in general use in France for aeronautic work and for many motor cars. For those who desire a mineral blend castor, the company advises the use of Wakefield's Castrol, Grade R.

The 1928 model Anzani Engines such as shown in section at Fig. 219 B, are especially adapted to American commercial use. They embody many new special features which assure the utmost in economy and dependability of operation for all requirements of from eighteen to 120 horsepower and the lubricating system has been changed to use a double gear pump. The principal reason for the unfailing performance of the Anzani lubricating system is its utmost simplicity—its few parts are rugged and easily accessible. From the tank the oil is led to a pressure pump which forces fresh oil supply to the main bearings and through holes and drilled passages in the crankshaft to the crankpins. The rotary force throws the oil in a fine mist which lubricates cylinder-walls, pistons and bearings. Likewise, the timing gears at the rear of the engine are amply lubricated by direct pressure supply. The excess oil collecting in the crankcase is drained through the scavenge line at the bottom of case. The oil before being put into circulation goes through a fine filter and, purified, is led back to the tank. Proper clearances are given all working parts in order to obtain best results from the use of mineral oil. All Anzani engines are tested with mineral oil (Mobiloil B). Any high grade oil corresponding to Liberty Aero Oil Numbers 2 and 3 may be used satisfactorily.

Efficiency of Oil Pumps.—Tests on the power consumption and volumetric efficiency of oil pumps as used on automobile and aircraft engines have been comparatively rare and American engineers may be interested in some tests of such pumps which were made in the Mechanical Laboratory of the Breslau Technical College, the results of which were briefly summarized in the *Zeitschrift des Vereines Deutscher Ingenieure.* The tests were made on a vane type pump from Hispano-Suiza aircraft engine and on two gear pumps from automobile engines. It was found that both the mechanical efficiency and the volumetric efficiency are dependent on the viscosity of the oil and on any leakage occurring. The vane type of pump has two chambers and the theoretical delivery per revolution is equal to two times the maximum volume of each chamber. Maximum delivery is assured when the inlet and outlet bosses are coaxial, 180 degrees apart, and perpendicular to the axis of eccentricity of the rotary piston. The theoretical delivery per revolution of a gear pump is equal to twice the amount of oil which is pressed out of the tooth spaces of one gear by the teeth of the other. In order to determine this it is necessary to measure the volume of the tooth spaces and of the teeth, and the clearance at the bottom of the tooth spaces must also be taken into account.

It was found from the results that the volumetric efficiency of the gear pump increases with the speed, while with the eccentric or vane pump the volumetric efficiency reached a maximum value at 1,500-2,000 r.p.m. The reason for this difference in the characteristics of the two types is said to reside in the oil, which, by reason of its viscosity, cannot follow the rapid impulses of the vanes at the higher speeds. With the gear pump the flow is more nearly uniform and the volumetric efficiency therefore continues to improve with increase in speed.

With gear pumps the volumetric efficiency decreases with the viscosity. This was also held true for the vane pump below 1,000 r.p.m., while at higher speeds a maximum volumetric efficiency was obtained for oil of a viscosity of three degrees Engler. Noticeable heating in the pumps occurred only when high viscosity oil was pumped against considerable pressure heads.

The plunger type of oil pump has been used to some extent abroad though most of the American aviation engines use either the vane or gear pumps to maintain oil circulation. If a very high pressure is to be maintained, plunger pumps are very satisfactory. The installation of the oil pump group in a Lorraine (French) engine is clearly shown in the longitudinal sectional view given at Fig. 220. The plungers are driven by eccentric and link motion from a short shaft driven by bevel gearing from the water pump drive shaft. Two scavenging plungers are used and one pressure plunger, all working in a three cylinder body member. The plunger type of pump is claimed to handle cold oil and oils of high viscosity better than vane or gear pumps. The construction can be clearly understood by studying the pump group outlined in the diagram which is a longitudinal section through a Lorraine twelve-cylinder engine.

Fresh Oil Systems.—All variations and combinations of the splash and of the force-feed lubricating-systems are classified under the term "crank-

case systems." The fresh-oil system differs fundamentally in that it feeds no appreciable surplus to the bearing surfaces, and the slight surplus that may be provided as a safety factor need not be recirculated. The tests involve two types of fresh-oil system; that is, the "full fresh-oil," providing for the lubrication of all bearing surfaces by small quantities of unused lubricant applied directly to the engine parts, and the "combination fresh-oil and crankcase system," the latter method furnishing fresh oil in minute quantities for cylinder lubrication and recirculated oil for the lubrication of bearings and other surfaces.

In the early stages of internal-combustion-engine development especially in motor cars the splash system of lubrication met the demands fairly well. The speeds were moderate, the fuels were of good quality and an ample supply of oil embodying the desirable qualities for proper lubrication was available. As the operating speeds were increased it became apparent that, to secure more positive lubrication of the rod and main bearings, some other means than the splash system was required, and the force-feed system gradually superseded the splash system. However, with the abandonment of the splash system, one marked advantage was sacrificed with it. With the splash system, an oily vapor was present in the crankcase which penetrated every part of the case, finding its way to the valve chambers, gear housing and pushrods, and it provided ample, in fact, often too much lubrication, for the cylinder-walls. The value of this oily vapor cannot be denied, as it was an ideal means of lubricating many parts of the engine.

With the advent of the force-feed system, engineers depended upon the throw-off from the crankshaft to lubricate the pistons and cylinders, with leads to auxiliary shafts and in some cases to the timing-gears. With this system, not nearly so much oil is in suspension to lubricate such parts of the engine as the push rods, camshaft and gears. Furthermore, the oil thrown-off from the shaft is in relatively heavy drops and is thrown against the interior walls of the crankcase in planes coincident usually with the crankshaft cheeks adjacent to the main bearings. It does not float around throughout the crankcase as does the oil that the dippers on the connecting-rods beat into a fine mist in the splash systems. In observing engines in operation with the splash and the force-feed systems, it has been noticed that a far greater amount of oil exists in suspension in the crankcase with the splash system. Some engines employing the force-feed system still retain the dip-trough beneath the connecting-rods for the sole purpose of producing the oily mist for lubrication of other parts of the engine and to eliminate any oiling difficulty that might result from tardy oil feed due to thickening at low temperatures when the engine is first started.

It is customary in some engines using large cylinders to use fresh oil for cylinder lubrication only and recirculated oil for the bearings, a combination that has produced some interesting results. When the full fresh-oil system is employed, the installation usually provides a mechanism for metering oil properly and forcing it through leads to the cylinder-barrels, main and crankpin bearings and timing-gears. The oil is injected into each cylinder at a point opposite the first bridge-wall of the piston at the down dead-center position of the piston. An oil-lead carries the lubricant to each

main bearing and to a centrifugal ring placed on the crankshaft cheek to throw the oil into the crankpin drilling through which it is carried to the crankpin bearing. The lubricator mechanism is driven from the camshaft so that it operates automatically with the engine speed. Such systems have been applied to very large engines used in automotive applications but aviation engines for the most part use the dry sump pressure system. The full fresh oil system is more economical of oil and as fresh oil is constantly supplied, there is but little danger of using oil contaminated by the fuel leakage, carbon or metal particles, in the bearings. Of course, filters used in aviation engines remove most of the foreign matter in suspension but these cannot stop or separate the volatile fuel that leaks past the rings when the engine is overprimed or that condenses when the engine cools down after stopping and which dilutes sump oil.

The methods of applying oil to an engine can be the cause of appreciable variations in the maximum power that the engine is capable of delivering. Consider table that follows, taken from the *S. A. E. Journal,* which gives the maximum-power readings of four engines, each of which was tested with two fundamentally different oiling-systems. One column presents the maximum-power readings when the crankcase system was in operation, and the other column reveals the increases in the maximum power when the fresh-oil system was installed. These increases were not caused by the same elements in all four engines, although over-lubrication can be held responsible as the underlying cause in each case. In Engines Nos. 1, 2 and 3, the crankcase system permitted too great quantities of oil to reach the combustion-chambers, in this way causing the oil to interfere with the fuel charge that the carburetor had metered out for the best full-load performance. Without question such interference does occur, and the fact is easily demonstrated in some engines.

MAXIMUM-POWER COMPARISONS

Engine Numbers	Test Numbers	Crankcase System, B. Hp.	Fresh-oil System, B. Hp.
1	70 and 71	25.10[a]	29.10
2	47 and 49	26.80	29.60
3	17 and 29	27.25	35.75
4	80L and 80B	31.90	36.11

In the case of each engine, the oiling system only was changed.

[a] The difference in power readings in the case of engine No. 1 was due to over-lubrication when the crankcase system was in operation.

Temperature Effect on Power Delivery.—L. H. Pomeroy, M.S.A.E., well-known internal combustion engineering authority, has carried out a series of tests which have been reported in the *S. A. E. Journal* to determine the effect of temperature of oil and water on power absorbed by friction. He first considers, very briefly, what causes mechanical friction in any given automotive engine, where the areas and the pressures are determined by the design, leaving the speed and the viscosity as variables. These are, first, the friction arising from the crankshaft, the camshaft and the connecting-rod bearings, which rotate; secondly, that of the pistons, piston

rings and the valves, which slide; and thirdly, that of the auxiliaries, such as the generator, the pump and the distributor. The first and the second of these are our immediate concern. The friction in the crankshaft, the camshaft and the connecting-rod bearings is, of course, of the well-known journal-friction type, differing, however, in that the loading and the direction of the loading are constantly changing. The second, piston friction, is interesting because it involves starting and stopping during each stroke, which may and probably do alter the character of the frictional resistance produced. The researches of the Lubrication Committee indicate that the friction of a flooded bearing is (*a*) proportional to the speed of the engine, (*b*) proportional to the area of the bearing, (*c*) independent of the pressure on the bearing, (*d*) proportional to the viscosity of the lubricant and (*e*) independent of the materials of which the opposing surfaces are composed.

It may be well to explain the term "friction mean effective pressure." During the last few years it has become a recognized method of criticism, when dealing with engine horsepower for various sizes of engines, to correlate them by referring to the mean effective pressure developed on the piston. In the working cycle, we have, of course, a negative pressure or suction during the inlet stroke and a positive pressure during the compression stroke, in each of which work is done upon the mixture; then the explosion stroke, when we take back at compound interest what we have lent during the compression stroke; and finally a positive pressure during the exhaust stroke when, having enfeebled our debtor all we can, we cast him into outer darkness. The average of all these pressures upon the piston is called the mean effective pressure. During the process described above, we must, of course, overcome the internal friction of the engine. This necessitates work, and this work can be related to pressure per square inch of piston-area in the same way that the fluid pressures are related. We then have a means of comparing the friction of various engines regardless of their size, number of cylinders or aught else. Frictional losses can be reduced by rational design but, even so, the effect of varying oil viscosity decidedly affects the engineer's best efforts, unless these include viscosity.

RESULTS OF FRICTION MEAN EFFECTIVE PRESSURE TESTS AT VARIOUS SPEEDS AND DIFFERENT OIL AND WATER TEMPERATURES

1000 R.P.M.

Friction Mean Effective Pressure, Lb. per Sq. In.	Temperatures, Degrees Fahrenheit Oil	Water
13.4	163	82
10.8	171	108
10.5	170	122
10.4	167	135
10.2	165	140
9.9	162	142
10.2	161	145
13.2	176	75
10.3	179	127
24.6	57	71
20.2	58	102
19.5	62	117
18.9	64	122

2000 R.P.M.

Friction Mean Effective Pressure, Lb. per Sq. In.	Temperatures, Degrees Fahrenheit Oil	Water
17.70	174	95
16.65	170	115
16.00	170	130
16.10	167	137
15.90	165	141
15.60	163	145
15.80	161	145
17.90	182	85
14.80	180	130
14.70	174	150
27.20	57	80
24.20	60	105
23.40	62	118
23.00	66	120

3000 R.P.M.

Friction Mean Effective Pressure, Lb. per Sq. In.	Temperatures, Degrees Fahrenheit Oil	Water
29.0	182	115
29.0	179	142
36.6	58	109
33.4	62	112
35.0	64	122

FUEL CONSUMPTION OF AN ENGINE RUNNING AT 1000 R.P.M. AND VARYING OIL AND WATER TEMPERATURES

Power Developed, hp.	3.40	4.70	4.95
Brake Mean Effective Pressure, lb. per sq. in.	10.70	14.70	15.60
Indicated Mean Effective Pressure, lb. per sq. in.	31.70	31.20	29.80
Mean Temp. of Oil and Water, deg. Fahr.	100	128	152
Fuel Consumption, lb. per b.hp-hr.	1.950	1.360	1.280
Fuel Consumption, lb. per i.hp-hr.	0.655	0.640	0.670

FUEL CONSUMPTION OF AN ENGINE RUNNING AT 1000 R.P.M. AND A CONSTANT THERMAL EFFICIENCY

Mean Temp. of Oil and Water, deg. Fahr.	100	128	152
Friction Mean Effective Pressure, lb. per sq. in.	17.0	12.5	10.2
Closed-Throttle Pumping-Loss, hp.	4.0	4.0	4.0
Brake Mean Effective Pressure, lb. per sq. in.	10.7	14.7	15.6
Indicated Mean Effective Pressure, lb. per sq. in.	31.7	31.2	29.8
Mechanical Efficiency, per cent	34.0	47.0	52.5

Summirg up Mr. Pomeroy states that we have, as between an engine working under cold and hot conditions, that is, over a mean oil and water temperature range of 100 to 150 degrees Fahrenheit, a reduction of 34.4 per cent in the gasoline consumption, as obtained from test-bench experiments under load, a reduction of 35.2 per cent indicated by friction measurements and, in the case of the actual operation of an automobile, a reduction of 33.8 per cent. It is freely granted that these are only isolated examples; nevertheless they bear an uncanny relation to one another and should not be dismissed without comment. The variation in body between

a thick and a thin oil at high temperatures may be of great importance even though the viscosity difference may be small. In brief, this small difference in viscosity is appropriate to a very large increase in the temperature of the heavier oil, since the viscosity-temperature curves of oils of greatly differing viscosity when cold become nearly parallel to the temperature axis. The engineer is, therefore, and not for the first time, on the horns of a dilemma. If he is working for a high efficiency under normal conditions, he is running risks at high temperatures, and vice-versa.

High Oil Outlet Temperature Not Always a Sign of Trouble.—A mistake often made by mechanics and field engineers is that high oil outlet temperature is a sign of trouble or poor engine design and this belief is fostered by the instruction given by engine builders in some cases. Mr. John H. Geisse of the Naval Aircraft Factory, Philadelphia, Pa., discusses this matter in *Aviation* and is authority for the statement that not only do oil temperatures have different values in different engines within a fairly wide range but that it may be different in engines of the same make to some extent without indicating serious trouble. Undoubtedly, there is a limit for each installation, but for different installations of the same engine, or for different engines, this limit may be anywhere from 120 degrees Fahrenheit to 240 degrees Fahrenheit, or even higher. The reason for this wide variation in permissible outlet temperatures is that the outlet temperature bears no direct relation to the temperature of the oil in the bearings, or to the heat generated in the bearings, the two values that do have a fairly definite permissible maximum. The analysis that is presented in the following paragraphs is intended to show why it is such a direct relationship does not exist.

To simplify the analysis, several assumptions will be made, which deviate fairly far from actual conditions existing in an engine, but Mr. Geisse believes that they will affect only the magnitude of influence of the various factors and not the direction of their influence. The major assumption is that the oil entering the engine is divided into three distinct channels which do not converge until they reach the scavenging pump. One channel will lead to the crankpin bearing, and from there to the pistons and cylinder walls, and will then contact with a section of the crankcase on its return to the scavenge pump. A second channel will lead to the auxiliary and main bearings, and contact with a different section of the case on its return. The third channel will lead through the relief valve and no heat will be added to, or subtracted, from this channel.

When the three channels converge, their temperatures are equalized and the mean thus obtained is the oil outlet temperature. The oil outlet temperature can then be set down in the following equation:

$$T_0 = \frac{W_1T_1 + W_2T_2 + W_3T_3}{W_1 + W_2 + W_3}$$

in which

T_0 = outlet temperature
W_1 = rate of flow in channel No. 1
T_1 = final outlet temperature channel No. 1

W_2 = rate of flow in channel No. 2
T_2 = final outlet temperature channel No. 2
W_3 = rate of flow in channel No. 3 (relief valve)
T_3 = inlet temperature

Now, let us consider the heat interchanges in channel No. 1. The rate of oil flow through the bearing will not materially change the friction, so it can be assumed that the amount of heat absorbed will be independent of oil flow. This being the case, the temperature of the oil leaving the bearing will be equal to the inlet temperature plus some constant, divided by the rate of flow. This oil will then strike the cylinder walls and pistons. Here, the amount of heat absorbed will increase with increase of oil flow at the expense of heat dissipated by the cylinder cooling means. We will make a rather bold assumption that the temperature of the oil leaving these parts will bear a definite relation to the metal temperature and, therefore, the heat units added to the oil in unit time will vary directly with the rate of flow. In passing over the crankcase wall, a certain amount of heat will be dissipated, and the amount will vary as the difference of the oil temperature (assumed constant) and the outside air temperature, the air velocity over the crankcase, the area wetted, and the rate of flow. However, in practically all engines, this loss of heat will not equal the gain. It is obvious, therefore, that there will be a temperature rise in this channel.

It should be noted here that the heat generated in the crankpin bearing will have very little direct effect on the final heat content in this channel. It will, however, have somewhat of an indirect effect, in that the reduction in viscosity of the oil associated with the higher temperature in the bearing will increase the rate of flow in this channel and therefore more heat will be taken from the pistons. It should also be noted that the temperature of the oil in the bearing is not related directly to the final temperature in this channel but does have a fairly close relationship to the oil inlet temperature.

Next consider the oil in the second channel. It likewise receives a definite amount of heat from the bearings, and the temperature leaving the bearings will be equal to the oil inlet temperature plus a constant divided by the rate of flow. This oil in passing over the crankcase will lose an amount of heat proportional to the difference in the temperature of the oil and the air, the wetted area, the air velocity, and the rate of oil flow. In any engine operating with no final increase in oil temperature, it is apparent that this oil must accomplish a total dissipation of heat equal to the amount added to the oil in the first channel, since the oil in the third channel has ro heat interchange.

Now let us consider the effect of various variables on the oil outlet temperature and the temperature rise. An increase in the rate of flow through the relief valve, without any other change, will reduce the outlet temperature without altering the temperature at the bearings. This is an important item. Engines having a low oil flow through the relief valve, such as the Liberty, will have a greater oil temperature rise than engines like the Wright T3, having a much larger relief flow, other things being equal. Engines like the LeBlond, in which the oil from the relief valve

goes back to the suction side of the pressure pump, giving no oil in channel No. 3, will have a greater temperature rise than will a Wright "Whirlwind," if other factors are alike. It is quite apparent then, that these engines will all require different temperatures of the oil outlet in order to secure the same temperature at the bearings.

Again let us assume two air-cooled engines alike in all respects, including total rate of oil flow, but with a different ratio of flows in channels No. 1 and No. 2. That one, having the greatest flow through channel No. 2 will have a lower temperature rise because channel No. 2 tends to dissipate heat and channel No. 1 to absorb heat. This is an item worthy of special consideration. In most installations of air-cooled engines, it is desirable to avoid the necessity of an oil cooler, depending on the radiation from the engine and tank to dissipate all of the heat. It is quite possible, that in cases where this has not been achieved, that a slight change in the oil system of the engine allowing a greater flow to the accessory drives would have accomplished the desired result. It is also possible that oil temperature regulation in air-cooled engines might be secured by regulation of the flow in the second channel, thus making adjustable cowling unnecessary.

Another point of interest in this division of flow is in the effect of detonation. Mr. Geisse is inclined to believe that the oil temperature is sometimes used as a guide in the choice of fuels for air-cooled engines. If this is the case, it is evident that that engine having the greatest flow to the pistons and cylinders will show the greatest increase in oil temperatures due to detonation and may possibly be considered as more inclined to detonation than an engine less sensitive. On the other hand the engine having the least sensitivity may be abused by the use of a fuel causing considerable detonation.

Now let us consider changes that may occur in an engine in service. In an air-cooled engine installation having no oil cooler, or other means of oil temperature regulation, an increase in bearing clearances in channel No. 2 will result in a decrease in oil temperature. An increase in the connecting-rod bearing clearance will result in just the opposite effect as explained in a preceding paragraph. An increase in oil pressure will increase the rate of flow in both channels No. 1 and No. 2. The result may be an increase, or a decrease, in temperature rise. To determine which would be the case, would require a closer analysis of the heat exchanges in these channels than Mr. Geisse has made. In water-cooled engines, it would without question result in an increase, the extent of change being dependent on the relative amount of heat normally taken from the bearings and the pistons. A decrease in viscosity of oil used will act in the same way as a change in pressure.

In engines in which the oil from the relief valve is led to the inlet of the pressure pump, it is quite possible that too low an inlet temperature will result in too high an outlet temperature. This may occur also in engines having pumps, whose capacity varies considerably with viscosity and pressure head. In either case, the decrease in flow will result in a decrease in total heat taken from the engine. If this decrease is not sufficient to compensate for the lower flow rate, there will be of necessity an increase in temperature rise. If the increase of temperature rise is more

than sufficient to balance the decrease in inlet temperature, then it readily will be seen that the outlet temperature will increase as the inlet temperature is decreased.

Mr. Geisse states that although his article has been drafted to show that the temperature of the oil at the bearings does not have a definite relation to oil outlet temperature, he does not contend that the outlet temperature has no significance. It has, very definitely, but not as much as some operators are inclined to believe and variations in it must be analyzed much more thoroughly than has been generally done. A thorough analysis may save a considerable amount of work and worry for operators of aircraft engines and a mechanic familiar with one engine should not judge another because its oil temperature is higher or lower in value than he is familiar with. He should get the information regarding oil temperature and pressures from authoritative sources.

Oil Temperature Control Not the Only Solution.—Mr. A. Ludlow Clayden, M.S.A.E., in discussing the remarks of Mr. Pomeroy brought out the fact that internal-combustion engine endurance and reliability depends so largely upon proper lubrication that the efforts of aviation engine designers to secure positive circulation of oil at the best temperature is highly commendable and that equal attention should be given to supplying only clean oil. With proper lubrication the life of the average engine would be doubled and probably quadrupled, because under the usual conditions the engine is lubricated only a part of the time. The function of the lubricant is not to take the place of cast iron, to make an octagonal bearing round, or to force its way to the right spot against resistance. Too great stress is laid on design; when cylinders are round and piston-rings fit perfectly, the amount of dilution is negligible, unless the manifold system is below the average. The function of oil is to provide a film upon which the piston and piston-rings may slide, not to prevent the escape of expanding gases by blocking up the cracks in the cylinder.

Not long ago the thermostatic control of water temperature was very important from the carburetion viewpoint; with modern manifolds it is much less important. Various authorities have brought out very clearly the substantial advantages that could be derived from the thermostatic control of oil temperature. Such control could consist of arranging either to heat oil in cold weather, or to cool it under summer conditions, or possibly a combination of the two. Oil-coolers are difficult things to construct. When anything like an ordinary radiator is used, the effect on the hot oil is to chill the film in direct contact with the surface that by its high viscosity and poor conductivity interferes very seriously with the cooling efficiency of a radiator. Heating it, on the other hand, is much easier to accomplish. The use of a comparatively heavy oil intended to be operated warm as compared with a comparatively thin oil intended to be operated cool, is that the accumulation of fuel will be less rapid in the former case.

The oil industry is now asked to produce lubricants that will work equally well hot or cold, pure or diluted, clean or dirty, evidently a somewhat difficult task. The first step is to produce engines that do not contaminate the oil with other fluids or solids. When this desirable stage of development has been reached, it will probably be possible to obtain enough

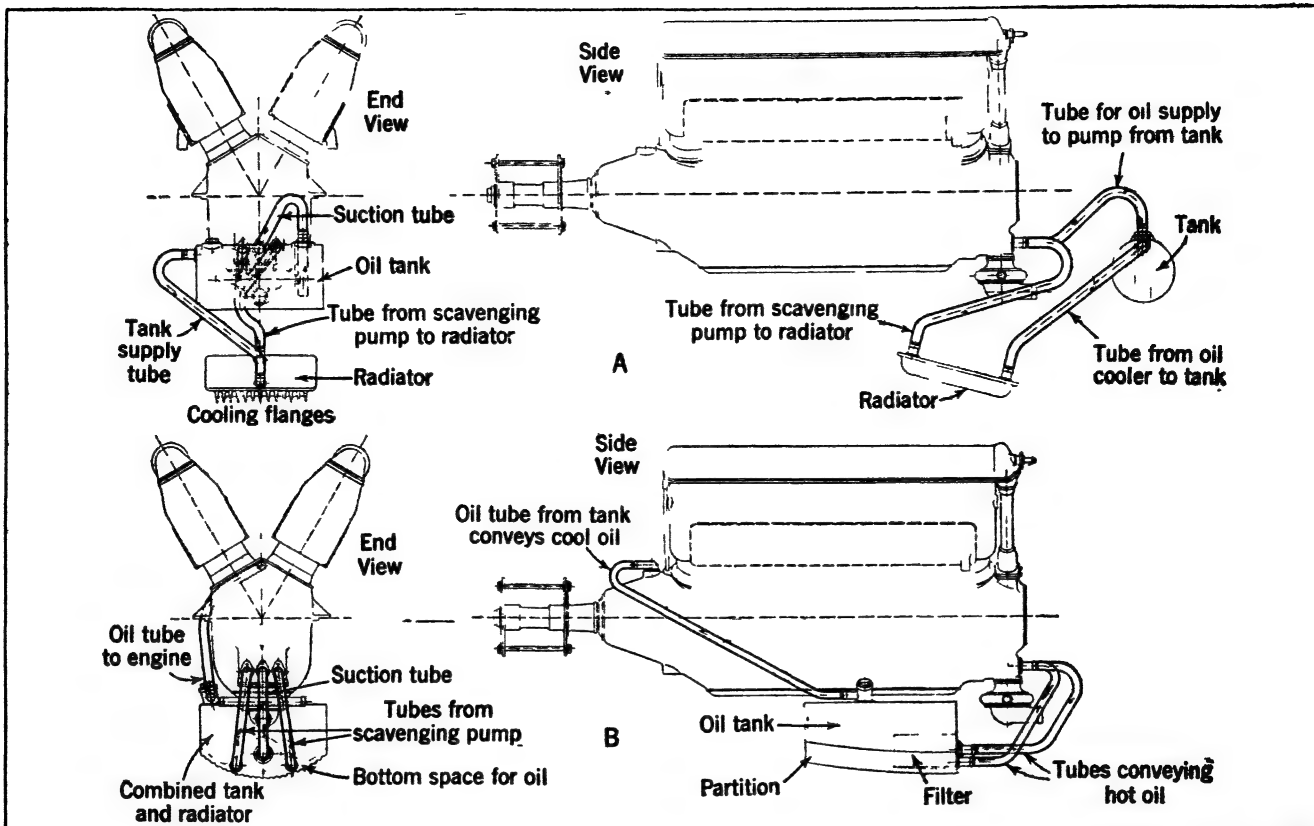


Fig. 221.—Methods of Installing Oil-Cooling Radiator Recommended by Hispano-Suiza Engineers. A—A Separate Oil Feed Tank is Employed, this Being Placed in Series with a Radiator. In the Method Shown at B the Radiator Forms the Bottom of the Oil Tank.

public interest so that lines of real research may be followed. Every engine manufacturer's instruction-book attempts to dodge this question, stating merely that oil should be changed at some definite mileage in the case of automobiles and after certain flying time in aviation engines. There are two objections to this method of dealing with the subject, the first being that the user does not always obey the instructions. It would be very surprising if he did, considering that more than 99 per cent of the automobile engines are designed so that it is impossible to drain the old oil without somebody's getting extremely dirty. Secondly, such draining very rarely removes more than one-half the accumulated solids that lie in the oil-pan, sump or crankcase and immediately contaminate the new oil which even frequent drainage, as in aviation engines, does not help materially to keep absolutely clean.

Hispano-Suiza Oil Cooling System.—Realizing the importance of maintaining the lubricating oil at the temperature that will best insure proper viscosity, engineers recommend that oil radiators or coolers be used in warm weather and oil heaters in cold weather. The application of an oil radiator or cooler which is separate from the oil supply tank is shown at Fig. 221 A as recommended by the Hispano-Suiza engineers. The use of a combined oil supply tank and radiator is shown at Fig. 221 B. It will be observed that the pipes carrying the scavenged oil discharge into the space at the bottom of the tank and all oil must pass over the air-cooled surface before flowing back into the supply portion of the reservoir from which it is drawn by the suction pump. The tank is divided into two compartments by a partition plate.

Wright Oil Temperature Control System.—Failures of the lubricating system have caused the loss of hundreds of valuable aviation engines, severe damage to the airplanes in which they were installed and have often placed the airplane personnel in great danger. These failures have frequently resulted from a lack of proper control of the oil temperature, so that the oil either became too cold to flow into the engine fast enough to give sufficient oil pressure, or else became so hot as to lose its effectiveness as a lubricant. Trouble due to cold oil has generally occurred in starting the engine in cold weather, and the usual means of overcoming it has been to drain the oil from the tank, heat it over a stove, and pour it back, an operation which is always inconvenient, and often impracticable. To keep the oil from becoming too hot, air-cooled oil radiators or water pipes in the oil tank have been used, but these have not been entirely satisfactory. After a long series of experiments, both in the dynamometer room and on airplanes in the field, the engineering department has developed an oil temperature regulator, which heats the oil rapidly in starting, and then holds its temperature at the proper point for good lubrication. Tests have demonstrated the effectiveness of this system both in hot and cold weather.

Although designed primarily for the control of oil temperature, this system has proved successful in the control of water temperature. In actual service it has replaced radiator shutter control with a resultant improvement in airplane performance, due to the elimination of the head resistance of the radiator shutters. An instance of this improved performance was

reported during the trials of the Navy SC1 Torpedo Scouting planes. These planes are fitted with particularly large radiators and it was found that with the shutters closed for cold weather operation, the closed shutters gave considerable drag. This was overcome by leaving the shutters open, using an antifreezing mixture and controlling the water temperature by the manually operated bypass which is a part of the Wright Oil Temperature Control System. With the shutters wide open, the pilot was able to control the temperature of the oil and water at will and the ceiling of the airplane was greatly improved. Acting on the evidence of these

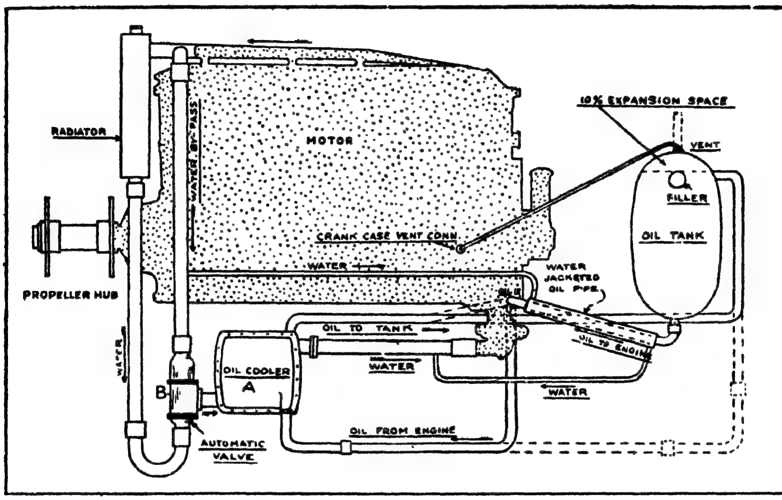


Fig. 222.—Oil-Cooling System Employed in Connection with Wright Water-Cooled Engines Utilizes Water Circulation of Engine.

trials the radiator shutters on these planes were removed; since the operation of the bypass valve is just as simple as the operation of the radiator shutters and has the advantage of giving better performance. A summary of the advantages follows:

(1) Sufficient oil pressure in starting to enable the engine to be idled safely, even in cold weather with cold oil.
(2) Full oil pressure in five to eight minutes, starting with cold oil.
(3) Reduction in time required to warm engine.
(4) Maintenance of oil temperature at proper point both in hot and cold weather.
(5) Elimination of necessity of draining oil tank overnight in cold weather.
(6) Ease of installation.
(7) Flexibility in design, allowing its use in airplanes already built.
(8) Simplicity in operation.
(9) Economy in oil.
(10) Protection of engine, airplane and personnel from a frequent and dangerous source of trouble.
(11) Correct engineering principles for maximum heat transfer per square foot of cooling surface.

(12) Eliminates necessity for pre-heating oil.

(13) Facilitates heating water in cold weather.

The Wright Oil Temperature Control System consists of a heat transfer unit or oil cooler shown at Fig. 222 A, a three-way valve (marked "B"), operated either by hand or by a thermostat, and the necessary water and oil piping. The heat transfer unit consists of a nest of thin copper tubes, with a shell and headers so arranged that the oil flows through the tubes, and the engine cooling water flows through the space between them. In this way the more viscous liquid has the path of lower resistance, and obstruction of oilflow is avoided. The headers are of cast aluminum alloy and are readily removable, and the whole unit is designed with ample strength to prevent distortion or leaks under service pressures. All the oil and water pass through the heat transfer unit at all times. The three-way valve controls a bypass around the main water radiator, so no water passes through the radiator until the water temperature has risen to its normal operating value. In this way the water temperature rises rapidly in starting and the oil is quickly warmed. When normal operating conditions are reached, the water keeps the oil cool. It is made in two sizes, one for engines ranging from 400 to 650 horsepower, the other for a range of 650 to 750 horsepower. The dimensions and weights are:

400-650 H.P. UNIT	650-750 H.P. UNIT
Overall dimensions—13" x 12⅞" x 8⅞".	Overall dimensions—13⅜" x 13" x 12¼".
Size of oil connections—¾" pipe tap.	Size of oil connections—¾" pipe tap.
Size of water connections—2" outside diam.	Size of water connections—2" outside diam.
Weight, empty—25 pounds.	Weight, empty—31 pounds.
Weight of contained water—10½ pounds.	Weight of contained water—14½ pounds.
Weight of contained oil—12½ pounds.	Weight of contained oil—13 pounds.
Cooling surface—32 sq. ft.	Cooling surface—50 sq. ft.
5-pass, giving equivalent tube length 45".	5-pass, giving equivalent tube length 45".

(a) Acting as a cooler, the 400 to 650 horsepower regulator is guaranteed to cool 5.5 gallons of oil per minute through twenty degrees Fahrenheit when supplied with 70 gallons of water per minute and when the temperature of the oil entering the cooler is not less than 35 degrees Fahrenheit above the temperature of the water entering the cooler.

(b) Acting as a heater, the 400 to 650 horsepower regulator is guaranteed to heat 5.5 gallons of oil per minute through twenty degrees Fahrenheit when supplied with 70 gallons of water per minute, provided the temperature of the oil entering the heater is at least 40 degrees Fahrenheit lower than the temperature of the water entering the heater.

EXAMPLE AS COOLER

Actual Tests

Wright T-3 Engine, 1,947 cu. in. displacement.

Average at 1,800 r. p. m., full throttle, 575 H.P.

Waterflow, 70 gals. per min.

Oilflow, 5.5 gals. per min.

Outlet water temperature, 180° F.

Inlet water temperature, 150° F.

Inlet oil temperature, 160° F.

Outlet oil temperature, 180° F.

Average oil pressure drop through oil radiator 0.3 lbs. per sq. in.

Average oil pressure in radiator:
Inlet 3.3 lbs. per sq. in.
Outlet 3.0 lbs. per sq. in.

The following data was obtained during three ground tests of planes equipped with the 400-650 horsepower oil temperature regulator:

EXAMPLES AS HEATER

Actual Tests on Three Different Planes

Date	Air Temp.	Oil Temp. At Start	Time Required to Obtain Initial Oil Pressure	Time Required to Obtain Full Oil Pressure	Oil Temp. at Instant Full Oil Pressure was Rep't'd
Jan. 4, 1924	45° F.	58° F.	Less than 1 min.	6 min.	72° F.
Jan. 24, 1924	40° F.	45° F.	Less than 1 min.	8 min.	102° F.
Mar. 10, 1924	44° F.	60° F.	Instantaneously	5 min.	87° F.

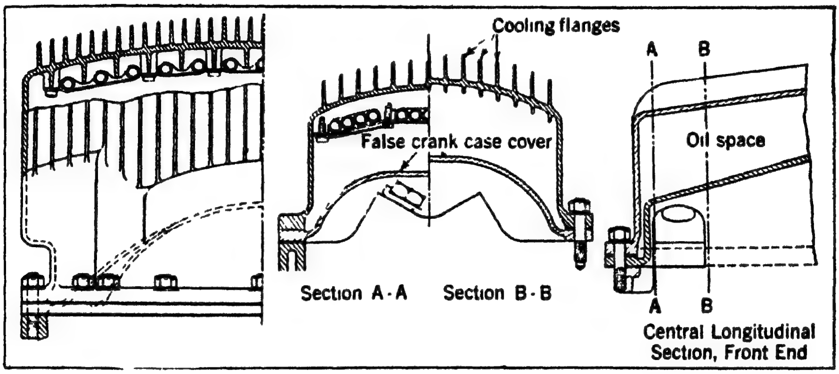


Fig. 223.—Detailed Views of Oil-Cooler of Some of the Packard Inverted Aircraft Engines.

Packard Oil Radiator.—Ever since the development of the water-cooled airplane engine with dry sump type of lubrication, defects in external oil lines, radiators, and tanks have been prime causes of engine troubles, resulting in forced landings. To eliminate these causes of trouble and thus make the aircraft engine more reliable an interesting self-contained oil cooler and reservoir has been developed by the Packard Motor Car Co., and was described and illustrated in *Automotive Industries.*

The new construction has been applied to the Packard 1,500 inverted aircraft engine. It consists of a false top for the crankcase. Above this false top there is an 8½-gallon oil reservoir. Inside this reservoir is a group of sherardized steel tubes longitudinally extending through it as shown at Fig. 223 which are fed from a main header at the accessories end of the engine. These tubes have holes drilled through their wall on top. Oil under pressure is forced into these tubes by the oil pump usually used to force oil through the oil radiator. The oil forced through the small holes in the top of the pipes is sprayed against the top of the oil reservoir, which is a finned aluminum alloy casting.

These fins project above the engine and take the place of the cowling formerly used with this type of engine. While the overall height of the

engine has been raised slightly through the addition of the false top to the crankcase and the fins, the additional drag is more than offset by the elimination of the oil radiator. The only external oil lines used are those from the pump to the oil pipe header. An internal standpipe has been included in the new design to take care of the overflow from the oil reservoir, this standpipe draining such oil to the timing gear case, which is capable of holding an additional 2½ gallons. A substantial saving in weight has also been achieved in this new design as compared with the previous engine with external oil supply system included.

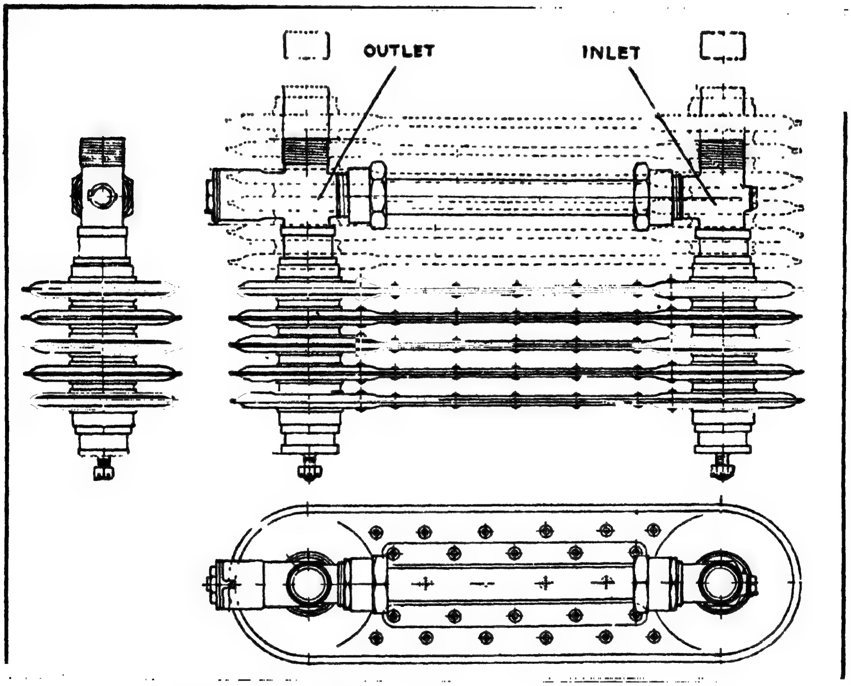


Fig. 223A.—Vickers-Potts Oil Cooling Unit.

The Vickers-Potts Oil Cooler.—In modern high-speed aircraft, with their high-powered engines cowled in as much as possible, the heat which is imparted to the lubricating oil during its passage through the engine can not readily be radiated from the crankcase—a system of cooling that gave more or less satisfactory results in the past. Today, therefore, it becomes necesary to seek some other means for reducing the temperature of the oil before it makes its journey through the engine and a number of such devices are described in this treatise. The usual method of doing this is to insert a special oil cooler in the pipe line between the engine scavenger pump and the oil tank. Such a device is the Vickers-Potts oil cooler, which forms the subject of the accompanying notes and illustrations, and which is manufactured by Vickers, Ltd., of Vickers' House, Broadway, Westminister, S.W.I., England.

The Vickers-Potts oil cooler—which is used extensively on machines of the Royal Air Force—is a standardized unit of comparatively low aerodynamic resistance, and can therefore be placed in the slipstream of the air screw or other convenient place, on practically every type of aircraft. It consists of a series of hollow fins threaded on two tubes, through which the oil passes on its way from the engine to the oil tank. These fins are arranged for series flow, i.e., through each fin or element in turn. A bypass valve is inserted between the inlet and outlet pipes, to provide an alternative path for the oil when starting from cold, and also to prevent excessive pressures on the fins. The internal construction of the cooling element

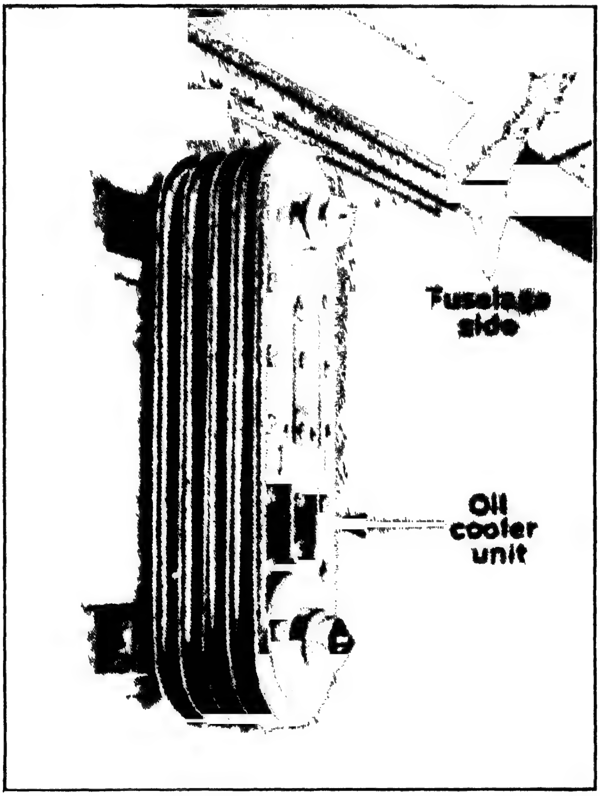


Fig. 223B.—Method of Installing a Vickers-Potts Oil Cooling Unit on the Side of the Fuselage.

is such that the oil is exposed in thin layers to the cold surface of the fins, while spacers between the fins break up the flow of oil by eddying, thereby causing a rapid transfer of heat. The external space between the fins—which is increased by the local flattening of the latter—enables the air to pass freely between them without causing undue drag.

These fins are all of standard dimensions, so that any number may be employed from five to eleven fins, to suit all engines, from 250 horsepower to 800 horsepower, and to meet the various requirements. Standard coolers are made with five, seven and eleven fins to cover this range. The cooling

surface per fin is approximately 145 square inches (930 square cubic millimeters), and the reduction of temperature of the oil in passing through one fin is from one degree to six degrees Centigrade, according to the rate of flow and the temperature of the air; the complete unit should permit oil being returned to the engine at a temperature of 70 degrees Centigrade. The complete weights, and the drag at 100 m.p.h. (air flow along major axis of fins) of the various units are as follows:—five-fin unit, weight 9.75

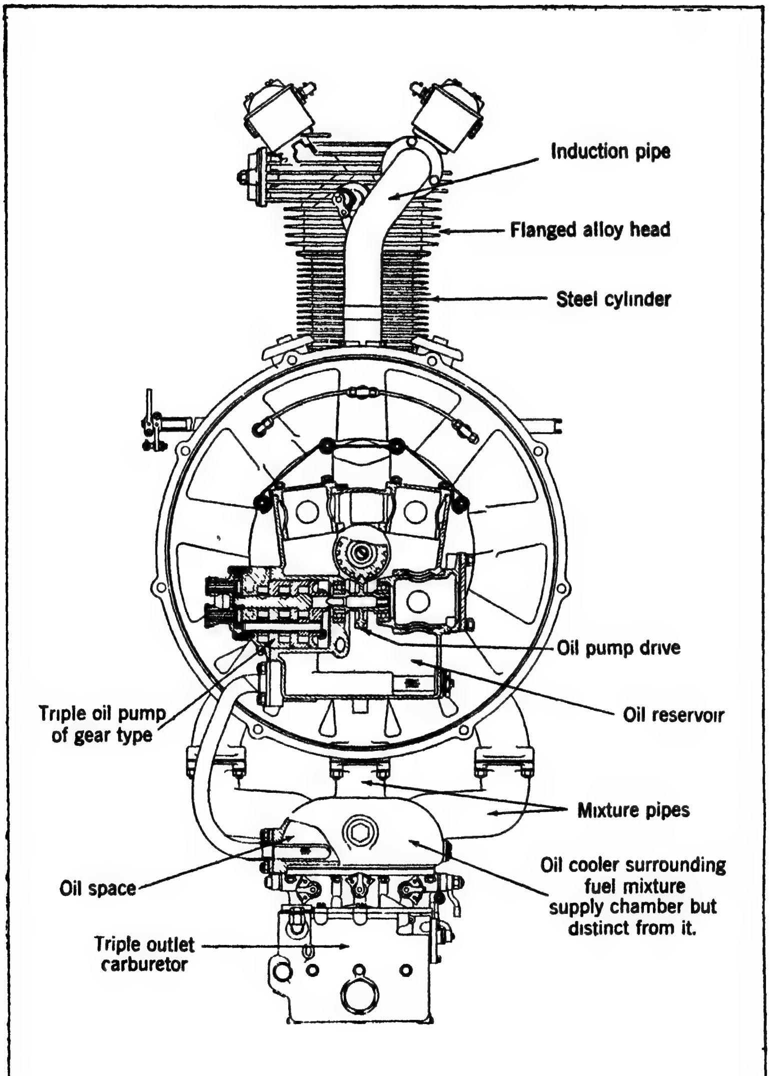


Fig. 224.—Diagram Showing Location and Method of Drive of Oil Pumps of the Wright "Whirlwind" Motor, and Use of Jacketed Induction Manifold for Cooling Oil.

pounds, (4.4 kgs.); drag, 1.16 horsepower; seven-fin unit, 11.75 pounds, (5.3 kgs.); 1.46 horsepower; nine-fin unit, fourteen pounds, (6.35 kgs.), 1.79 horsepower; eleven-fin unit, 16.5 pounds, (7.5 kgs.), 2.1 horsepower. This cooler measures approximately one foot four inches in length by four and one-half inches wide, and projects from the fuselage, etc., from about five inches in the five-fin unit, to about nine and three-quarter inches in the eleven-fin unit. The construction of the five section cooler is shown at Fig. 223 A and its installation at the side of the fuselage where it projects into the air stream is shown at Fig. 223 B.

Oil Cooling by Intake Gas.—In air-cooled radial engines the problem resolves itself into that of cooling the lubricating oil as the higher operating temperatures and the rapid attainment of efficient operating temperatures soon after starting do not call for oil heating means as may be necessary with water-cooled engines. As authorities recommend that a start in cold weather be made only with a fresh charge of heated oil, the system having been drained when the engine was stopped previously, it will be evident that preheated oil is available at once and a very few minutes engine operating time suffices to thoroughly heat all parts of the mechanism. Satisfactory oil cooling may be obtained by surrounding the mixture header or induction manifold with a jacket through which the oil is circulated as shown in Figs. 208 and 224 which outline the method of oil cooling and mixture heating used by the Wright engineers for Whirlwind engines. Considerable heat will be absorbed from the oil by the vaporization of the liquid particles of fuel in the air passing through the carburetor, because vaporization has a refrigerating action.

Ball and Roller Bearings Have Little Friction.—It is for this reason that many engineers use anti-friction bearings of the ball or roller types in important and heavily loaded bearing points of aviation engines. The diagram at Fig. 225 shows a longitudinal sectional view of the Napier-Lion twelve-cylinder W type water-cooled engine. This is useful in showing the engine oiling system as well as the application of roller bearings to the engine crankshaft. Five anti-friction bearings carry the main loads and while supplementary plain bearings are carried at the ends of the shaft, their function is primarily that of oil distributing and vibration dampening members as the roller bearings have more than ample capacity to carry the loads produced by the explosions. The propeller shaft drive is by gearing and this shaft is also supported by roller bearings to resist radial loads and a ball thrust bearing to resist the reaction of the propeller pull. Anti-friction bearings are also widely employed in radial cylinder engines, in some applications they are mounted even in the crankpin end of the master rod as in Siemans-Halske engines.

In the oil-lubricated journal-bearing lower running friction coefficients can be obtained, under certain conditions, than can be found in the best of ball-bearings. The friction in an oil-lubricated journal at starting, before the complete film is formed, is very high, running up to fifteen or twenty per cent. But after the start the friction falls off very rapidly, and the coefficient for moderate speed falls very low, below that of ball- or roller-bearings. As the speed is further increased, the journal-bearing friction increases more rapidly than that of ball-bearings, soon equaling it

and then going higher. The *durability* of the oil-film journal-bearing at high speed is said to offset any disadvantage due to higher friction but it has been the writer's experience, based on the oversight of several thousand engines operated under service conditions during the World War, these using both plain journals and anti-friction bearings, that the latter last longer than plain bearings do in similar applications and are more easily serviced in the rare instances where they gave trouble.

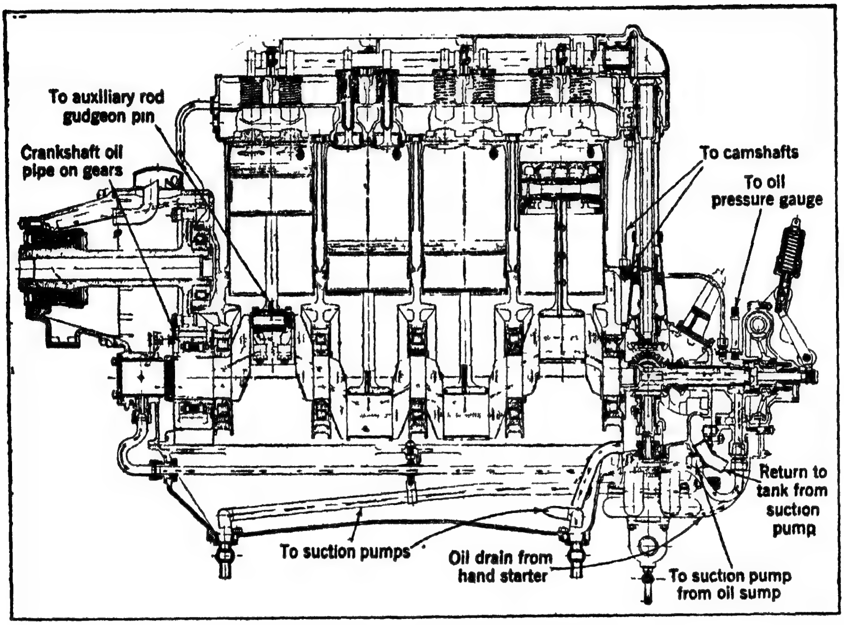


Fig. 225.—Sectional Diagram Showing Napier-Lion Aviation Engine with Internal Oil Piping and the Use of Anti-Friction Bearings for Crankshaft Support.

Mention has been made of oil viscosity, showing that it is a direct factor in the friction of journal-bearings. There has been much speculation as to how thin the oil-film can be and have the frictional resistance to rotation still follow the direct factorial relation set forth by Kingsbury. In 1901 Mr. Kingsbury measured the friction of oil-films in his Taper-Plug Viscosimeter and found that, at constant temperature and speed, the friction was inversely proportional to the film thickness down to the least thickness that he could measure, which was about 0.000025 inch. The law was followed quite definitely for thickness variations of 0.000001 inch. To measure such small amounts seems almost incredible. The plug was supported on the point of a screw to which was fastened a long lever whose outer end, when moved just a little, would cause the film to increase or decrease by increments of 0.000001 inch, (one millionth of an inch). These thickness variations produced a corresponding fall or rise in the friction. The results when plotted fell almost exactly on a fair curve.

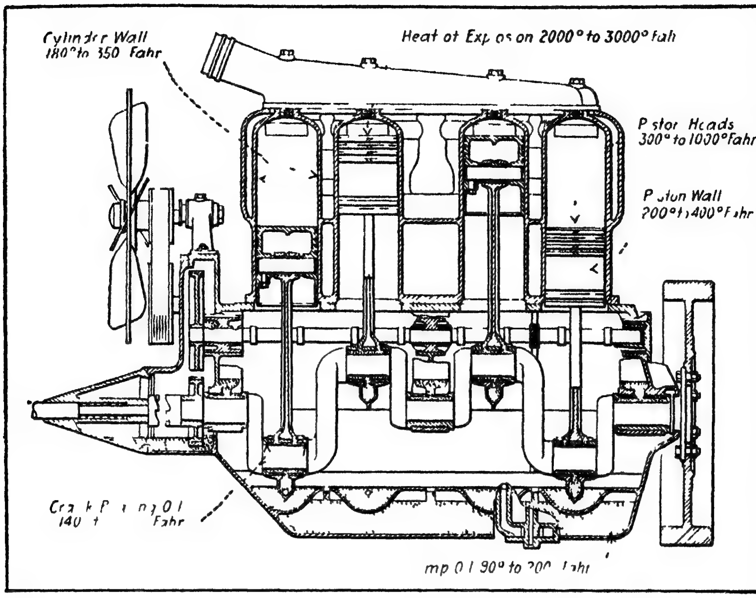


Fig. 226.—Operating Temperature of Automotive Engine Parts Useful as a Guide to Understand How Some Parts of Airplane Powerplants Heat Up More than Others.

QUESTIONS FOR REVIEW

1. Why do aircraft engines present a difficult lubrication problem?
2. What is the effect of varying bearing clearance on lubrication of engines?
3. Describe oiling system of Wasp engines.
4. Outline method of oiling "Whirlwind" engines.
5. Describe Liberty 12 oiling system.
6. How are Anzani engines oiled?
7. What factors control oil pump efficiency?
8. What is the temperature effect on power delivery?
9. Describe practical methods of oil temperature control.
10. Is high oil temperature always a sign of trouble?

CHAPTER XVIII

AIRCRAFT ENGINE COOLING SYSTEMS

Why Cooling Systems Are Necessary—Temperature of Engine Parts—Air-Cooled Engine Temperature—Reducing Back Pressure—Air-Cooled Engine Development—Air-Cooling Efficiency—Radial Cylinder Placing Ideal for Air-Cooling—Vee Type Air-Cooled Engines—Limitations to Air-Cooling Possible—High Engine Speeds Favor Water-Cooling—Cooling Systems Generally Applied—Cooling by Positive Water Circulation—Water Circulation by Natural System—Radiator Location—Resistance of Radiators—Direct Air-Cooling Methods—Air-Cooled Engine Design Considerations—Air-Cooling Permits Important Weight Saving—Experience of U. S. Navy with Air-Cooling—Air-Cooled Engines in Pursuit Planes.

The reader should understand from preceding chapters that the power of an internal-combustion motor is obtained by the rapid combustion and consequent expansion of some inflammable gas. The operation in brief is that when air or any other gas or vapor is heated, it will expand and that if this gas is confined in a space which will not permit expansion, pressure will be exerted against all sides of the containing chamber. The more a gas is heated, the more pressure it will exert upon the walls of the combustion-chamber in which it is confined. Pressure in a gas may be created by increasing its temperature and inversely heat may be created by pressure. When a gas is compressed its total volume is reduced and the temperature is augmented.

Why Cooling Systems Are Necessary.—The efficiency of any form of heat engine is determined by the power obtained from a certain fuel consumption. A definite amount of energy will be liberated in the form of heat when a pound of any fuel is burned. The efficiency of any heat engine is proportional to the power developed from a definite quantity of fuel with the least loss of thermal units. If the greater proportion of the heat units derived by burning the explosive mixture could be utilized in doing useful work, the efficiency of the gasoline engine would be much greater than that of any other form of generating power by combustion. There is a great loss of heat from various causes, among which can be cited the reduction of pressure through cooling the motor and the loss of heat through the exhaust valves when the burned gases are expelled from the cylinder.

The loss through the water jacket of the average internal-combustion automotive powerplant is over 50 per cent of the total fuel efficiency. This means that more than half of the heat units available for power are absorbed and dissipated by the cooling water. Another sixteen per cent is lost through the exhaust and less than $33\frac{1}{3}$ per cent of the heat units do useful work because there are other losses to be considered as well. The great loss of heat through the cooling systems cannot be avoided, as some method must be provided to keep the temperature of the engine within proper bounds. It is apparent that the rapid combustion and continued series of explosions would soon heat the metal portions of the engine to a red heat if some means were not taken to conduct much of this heat away. The

high temperature of the parts would burn the lubricating oil, even that of the best quality, and the piston and rings would expand to such a degree, especially when deprived of oil, that they would seize in the cylinder. This would score the walls, and the friction which ensued would bind the parts so tightly that the piston would stick, bearings would be burned out, the valves would warp, and the engine would soon become inoperative.

Temperature of Engine Parts.—The best temperature to secure efficient operation is one on which considerable difference of opinion exists among engineers. The fact that the efficiency of an engine is dependent upon the ratio of heat converted into useful work compared to that generated by the

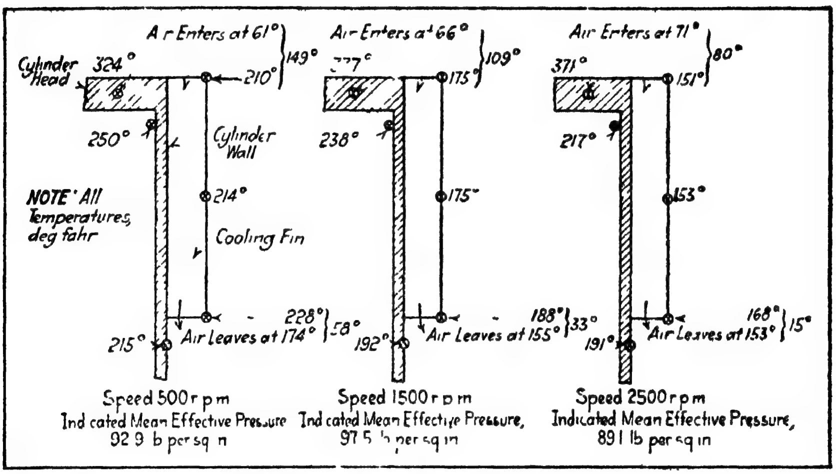


Fig. 227.—Cylinder and Fin Temperature in an Air-Cooled Automobile Engine. All Measurements are in Degrees Fahrenheit, and Were Made at Wide Open Throttle with No Auxiliary Cooling at Speeds Ranging from 500 to 2,500 R.P.M. Temperatures Were Obtained by Thermo-Couples and Potentiometer System. Note that in Every Case the Hottest Point was Between the Sparkplug and the Exhaust Valve.

explosion of the gas is an accepted fact. It is very important that the engine should not get too hot, and on the other hand it is equally vital that the cylinders be not robbed of too much heat. The object of cylinder cooling is to keep the temperature of the cylinder below the danger point, but at the same time to have it as high as possible to secure maximum power from the gas burned. The usual operating temperatures of a water-cooled automobile engine are shown at Fig. 226, and this can be taken as an approximation of the temperatures apt to exist in an airplane engine of conventional design as well when at ground level or not very high in the air. The newer very high compression airplane engines in which compressions of six or seven atmospheres are used, or about 125 pounds per square inch, will run considerably hotter than the temperatures indicated.

Excessive back-pressure in exhaust-manifolds and mufflers is a fault to be found in many modern motor cars. Resultant overheating of engines, valves and pistons may cause serious trouble which can be avoided with a little study, according to C. P. Grimes, research engineer of the H. H. Franklin Mfg. Co. Mr. Grimes gave an extremely interesting talk before

the Indiana Section of the S. A. E., presenting in an informal way many practical developments and conclusions based on experiments in the Franklin laboratories. Engine-testing procedure has been placed on a scientific plane in the Franklin experimental shops. Before starting a test, the engine is torn down and inspected thoroughly to see that the dimensions and the tolerances conform with the drawings. After reassembling, the engine is run for a few days to overcome initial tightness and any excessive friction. Combustion-chambers are each checked for cubical contents by measuring the amount of oil that can be held in each. The compression is checked with an O'Kill indicator. A neon-filled tube is so arranged in the ignition circuit that it projects a spot of light onto a polished brass disc which rotates on the crankshaft and thus the spark-timing can be checked accurately and visibly. Mr. Grimes emphasized strongly the value of the Leeds & Northrup potentiometer in engine testing, saying that he could not do satisfactory work without it. As an indication of the extreme care taken to secure accurate results, Mr. Grimes cited that friction tests of the engine are taken frequently throughout the period of any run to be sure that wear or improper lubrication has not introduced an error that might lead to wrong conclusions.

Air-Cooled Engine Temperatures.—In the development and refinement of air-cooled engines, the Franklin organization has made many measurements of air and engine temperatures. The temperatures prevailing in the flywheel fan design of engine are shown diagrammatically in Fig. 227. Mr. Grimes called attention to the fact that there is a considerable rise in temperature of the air between the points where it enters the cylinder fins and where it leaves them. This temperature rise is from two to three times that effected when air passes through the average radiator of a water-cooled engine. This better performance is due to the use of an efficient exhaust-fan and housing as opposed to the customary blade fan blanketed behind a radiator and in front of the cylinder block. Scientific design has reduced the power consumption of the cooling fan to 0.32 horsepower at ten m.p.h. and 1.30 horsepower at 30 m.p.h. (In air-cooled aviation engines there is no power loss due to fan or blower because the propeller slipstream is utilized in cooling the cylinders which project into the air stream.)

Efforts to reduce engine temperatures to a minimum led the Franklin engineers to investigate the effect of exhaust back-pressure. It was found that the dual type of exhaust-manifold was of material advantage in reducing back-pressure, but even more effective means were developed. Each drop in back-pressure effected an increase in torque and power. Engine temperatures were reduced materially. Mr. Grimes said that some cars are produced today with manifolds that cause a back-pressure of from eight to fourteen inches of mercury. The present Franklin design reduces back-pressure to two inches of mercury, and maximum exhaust gas temperatures have dropped from 520 to 260 degrees Fahrenheit. Stoves for heating intake air must be increased in size with each reduction in back-pressure since the exhaust is running cooler. This matter is of interest at a time when aviation engineers are experimenting to provide effective manifolds and mufflers for aircraft engines.

Reducing Back-Pressure.—Properly designed exhaust rings or collecting manifolds are a material factor in the elimination of back-pressure as well as silencing the exhaust. Strange as it may seem, experiments made some years ago served to indicate that the exhaust through a properly designed manifold might offer considerably less back-pressure than a direct exhaust from the ports to the air. In the writer's opinion, the reason for this is that the gas stream flowing from the exhaust ports has a definite

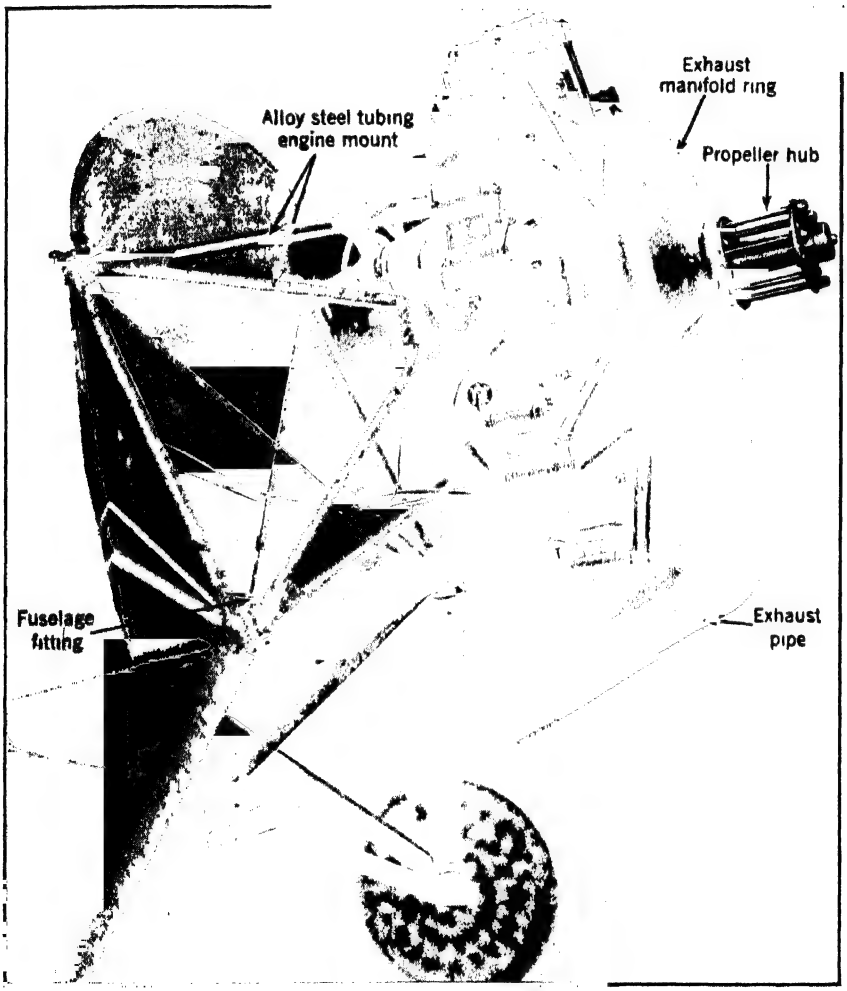


Fig. 228.—Installation of Aircraft Engine of Siemens Design on American Eagle Airplane Fuselage Showing Method of Installing Exhaust Gas Collector Ring.

area and of course, a pressure three or four times that of the air. The resulting reaction can be understood by the somewhat crude analogy of slapping water with a board quickly and feeling the resistance as compared with immersing the board more gradually. When exhaust gas issues through a manifold or even a series of properly designed stacks it issues

to the air at less pressure and greater volume and consequently meets with less resistance.

The exhaust ring provided as standard equipment on Ryan-Siemens engines, distributed by the Ryan Aeronautical Corp., San Diego, Calif., is in front of the cylinders as shown at Fig. 228 fairing into the cowling on the plane and providing the proper streamlining in front of the cylinders to give even cooling. It consists of an annular ring with vents connected to the exhaust ports of the cylinders. The gases leave at the bottom of the annular ring through pipe carried below the engine. This is surrounded by a hot-air stove with a screened opening, which is the carburetor intake, facing forward giving a slight amount of supercharging effect. There is an adjustment above the stove to allow a regulation for the amount of air being drawn in by the carburetor. The construction of the manifold when removed from the engine is clearly shown at Fig. 229.

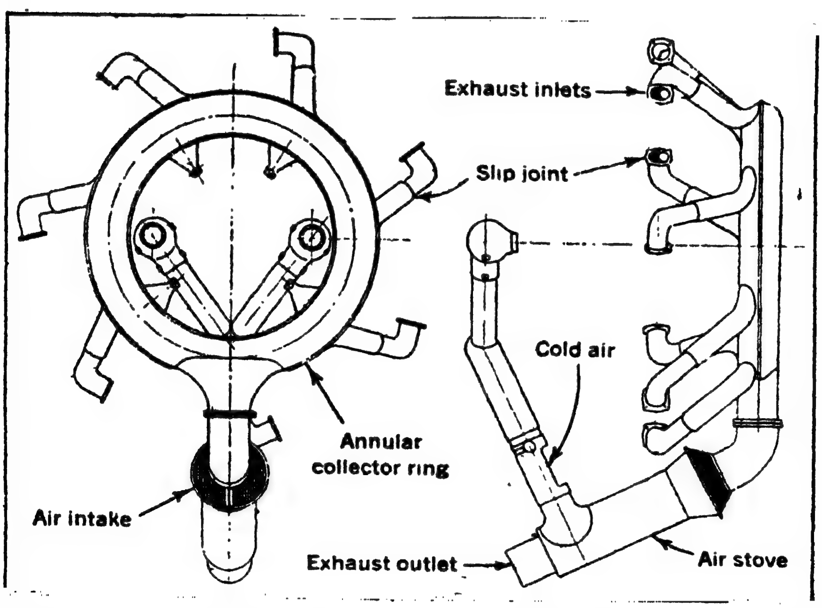


Fig. 229.—Drawing of Complete Exhaust Manifold System of 100 Horsepower Seven-Cylinder Siemens Engines. Note the Heater Stove and the Connection to the Carburetor Fitted with a Cold Air Bypass.

The system is said to reduce the noise and because of its arrangement reduces the fire hazard. It is made up of thin sheet iron with the annular ring in two halves stamped out with the seams welded together. The individual exhaust pipes are welded to the ring and a slip joint is provided to allow for expansion and contraction. The complete system, with preheaters, weighs thirteen pounds, sixteen pounds, and twenty pounds for the five-, seven-, and nine-cylinder engines respectively, and offers very little back-pressure, besides giving the important advantage of preheating the air entering the carburetor by combining an air stove with the exhaust gas discharge pipe.

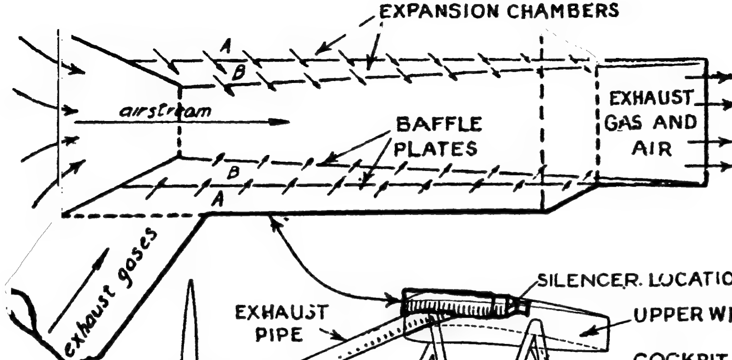
Elimination of noise in airplanes seems to be a general tendency in latest design. The inconveniences of the roar of the powerplant are apparent and it is natural that one's attention should be directed to the principal source of noise—the exhaust. Of course, the sounds given off by the propeller, as well as that of the engine itself, are considerable factors, but it would be useless to attempt to eliminate these until the exhaust of the engine has been quieted. Considerable effort, both in this country and abroad, has been directed to the silencing of engine exhaust noises. Abroad, mufflers are more common than in the United States where long exhaust pipes or exhaust manifolds with "silencing" ends are increasing in use. This subject was considered in detail by Mr. Richard M. Mock, writing in *Aviation,* and excerps which follow are taken from that source.

The loss of engine power, and also the tendency to burn valves when operating in hot climates, through increased back pressure in the exhaust muffler has been one of the "bugbears" of the engineer. However, it has been demonstrated that, with proper design, power loss may be decreased to a minimum, or, in some cases, the engine power increased. In addition, an exhaust silencer, of good design, has the advantage of eliminating the flames of the burning gases from the view of the pilot for night flying. Objections have been raised because of the increased weight and added air resistance, but with present day refinements in airplane design these objections have been reduced to the point where they are overshadowed by the advantages in the reduction in sound.

The standard aircraft engine, on most production planes, has an exhaust manifold which shows little effort to silence the exhaust. The most common type of manifold, on radial engines, is a ring for collecting the exhaust gases and allowing these to escape through vents at the bottom. The engine with the cylinders in line usually has a common collector manifold mounted on each bank of cylinders, with an opening at one end. On some designs there are short stacks on each cylinder or a group of cylinders. Each type of exhaust produces a different type of noise, and from casual observation it does not appear that any consideration of the elimination of noise has entered into most manifold designs. Most American engine manufacturers do not supply exhaust manifolds but leave their construction to the airplane manufacturer. An exception to this should be noted in case of the Wright J6 series which have a well designed exhaust ring incorporated on each engine.

It appears that the ultimate muffler will be of the Venturi type or long exhaust pipe type fitted with special ends. (These seem to be the only types which silence the engines sufficiently to warrant their use.) The resistance of the long exhaust pipe is slight when compared with that of the Venturi. It is true that the Venturi type can increase the speed of the engine but this power is absorbed from the airstream and therefore whether or not it does increase the overall efficiency of the installation is questionable. The decrease in pressure in the exhaust manifold increases the rate of exit of the exhaust gases appreciably.

Noises can be eliminated from an engine exhaust by slowly reducing the velocity of the burned gases. This may be accomplished by three methods:—regular expansion, change of direction, or surface friction.



Diagrammatic sketch of the exhaust silencer on a Loening cabin amphibian. The exhaust manifold is connected to one side of the silencer so that the gases swirl as they expand passing from chamber (A) to chamber (B) and then into the airstream.

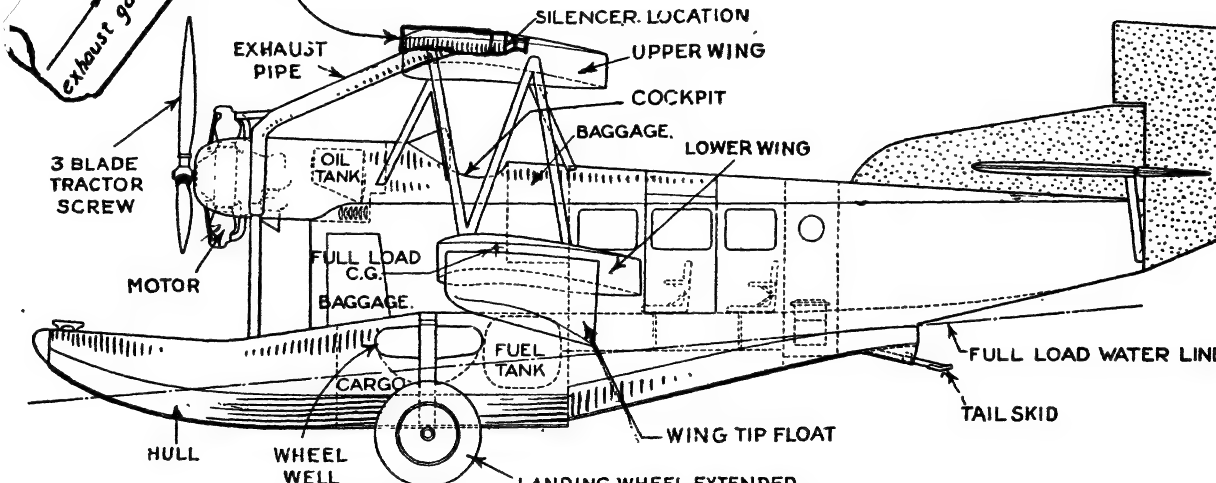


Fig. 229A.—Sketches Showing Construction of Exhaust Silencer Experimented with by Loening and Method of Installation on Amphibian Airplane.

Most exhaust silencers employ a combination of two or three of these. The simplest type of silencer is the long exhaust pipe, common on some Fokker planes. The exhaust manifold is connected to a flexible tube suspended longitudinally below the fuselage extending back to a point under the cabin. The gases expend gradually as they approach the end of the tube and their velocity is decreased by the friction of the walls of the tube. This method causes only a slight back pressure and the noise can be directed away from the cabin. The usual type of cockpit heater reduces the noise slightly by this method as it requires a somewhat longer exhaust pipe than is normally required.

The method of changing the direction of the gases and thus reducing their velocity, such as by the use of baffle plates as on automobile mufflers, causes too great a reduction in power to warrant its use on aircraft engines. In addition, the installation is usually quite heavy. Another method of changing the direction of the gases and thus slowly dissipate their energy is that of the exhaust manifold with the so-called "silencing" ends. The end of the exhaust pipe is either completely sealed or partly closed, while the wall is perforated with small holes having an aggregate opening more than sufficient to exceed the area of the pipe interior.

The general tendency in the installation of exhaust systems is to have an exhaust below the fuselage. This is due to the increasing use of cabin heaters and the desire to keep the exhaust flames out of view and to carry the exhaust noises as far from the cabin as possible. A muffling end on this pipe would decrease the noise to a minimum and eliminate the flame.

The muffler installed in the Loening commercial amphibian is of the Venturi type combined with the whirl type. A single muffler is used, connected to the top of the exhaust ring behind the cylinders of the Wasp engine. One is used for all the cylinders. The installation weighs 95 pounds complete with exhaust manifolding. It should be noted that the muffler is some distance from the engine, above the upper wing, and is connected by a large pipe. The exhaust enters by small holes and the shape of the Venturi has been simplified. The exhaust manifold, instead of being connected to the expansion chamber concentrically, is connected on one side so that the gases enter tangently and swirl in the expansion chamber. A baffle in the expansion chamber also tends to reduce their velocity. On the ground, with the muffler in the slipstream, it increased the speed of the Pratt & Whitney 400 horsepower Wasp engine from 1,630 to 1,660 r.p.m. The construction and installation of this muffling device is clearly shown at Fig. 229 A.

Bristol Jupiter Exhaust System.—For the assistance of aircraft constructors it is the policy of the Bristol Company to develop, test and standardize for each type of engine a suitable exhaust system, to act as an exhaust collector, flame damper and silencer. The type evolved for the geared Jupiter engines incorporates several novel features, the utility and effectiveness of which have been proved by extended bench and flight tests. The deflector ring incorporated serves the dual purposes of preventing heat radiation from the exhaust ring to the gear case, and also preventing hot spots and stagnant areas, by deflecting a steady flow of cooling air around the rear of the ring. Special spherical assembly joints and sliding expan-

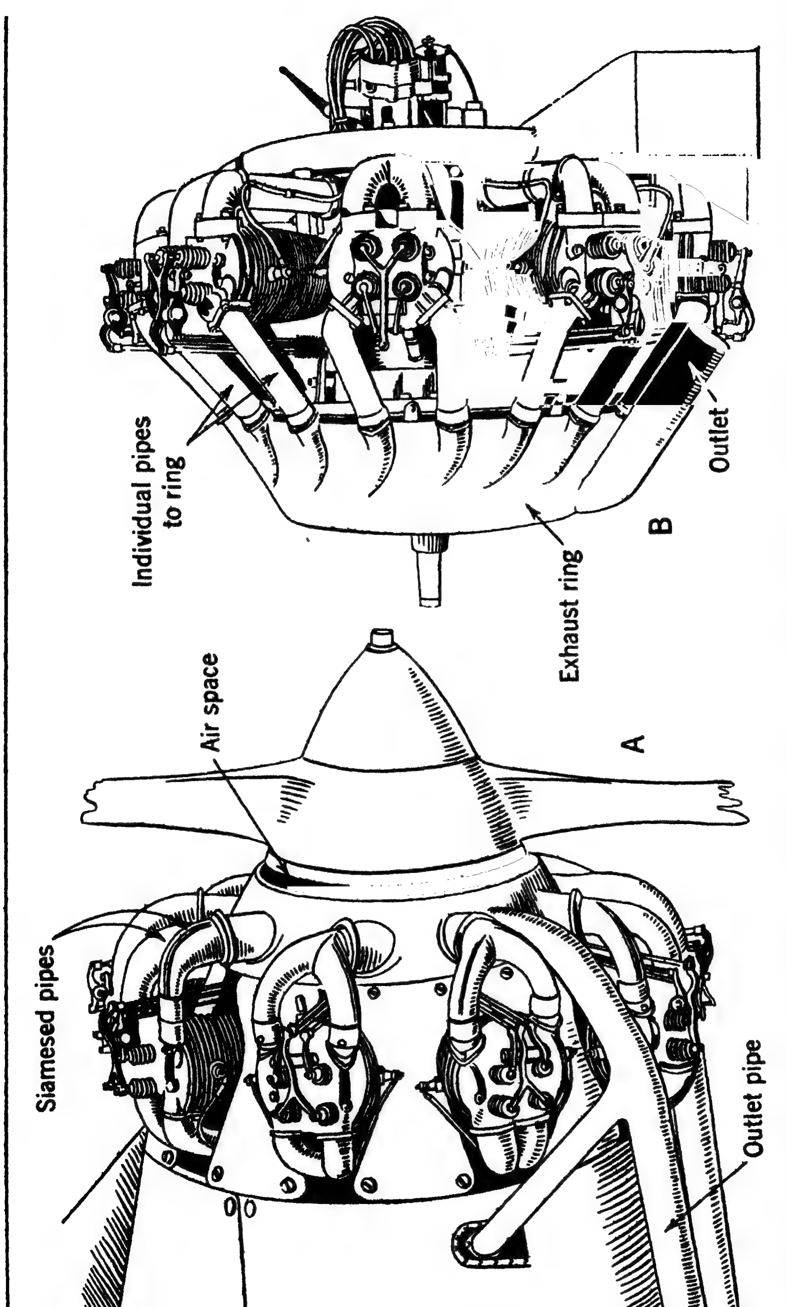


Fig. 229B.—Diagrams Showing Application of Bristol-Jupiter Exhaust System. A—Showing Cowling used when Engine is Installed in Fuselage. B—View of Engine Removed from Fuselage Showing Connections to Exhaust Ring.

sion joints are provided to ensure easy accessibility with interchangeability, and freedom from expansion and contraction stresses under running conditions. The ring is completely supported from special lugs on the engine gear case, and attachments are provided to allow of it being used as a front support of the cowling. This complete system has been fully tested and the detail design and dimensions are such that back pressure and consequent drop in power is negligible, being approximately one per cent. The rings are finished with the Fescolized process, making them proof against corrosion, with the result that the life of the rings can be considered as equal to that of the engine.

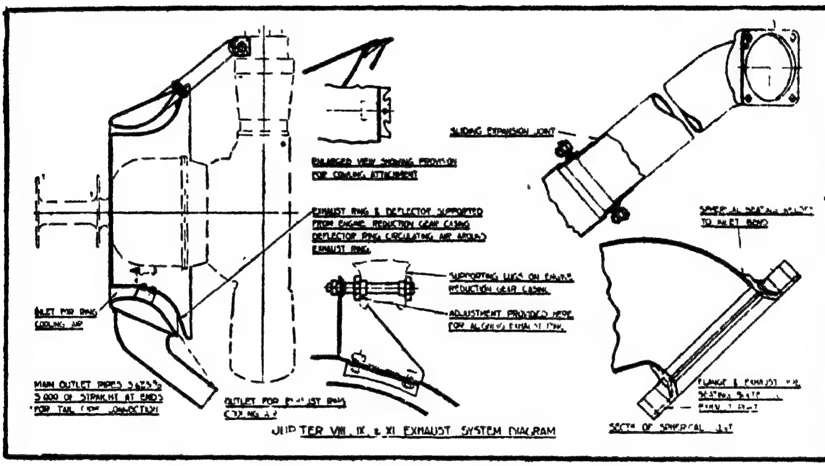


Fig. 229C.—Diagram of Jupiter Exhaust System.

There are two methods of joining the exhaust ports, of which, there are two per cylinder to the exhaust ring. The one shown at A Fig. 229 A "siameses" the two exhaust pipes to one member fastened to the ring, the other method is to use individual pipes to the ring, as shown at B Fig. 229 A. The general features of installation are clearly shown in the drawing at Fig. 229 C which is self-explanatory.

Air-Cooled Engine Development.—Older types of air-cooled engine suffered somewhat in comparison with water-cooled engines in economy, as cylinder designs were so faulty as to require over-rich mixtures to prevent detonation and preignition in high-powered engines. In the Model J5 engine and in the Pratt & Whitney Wasp engine, the cylinder design has been greatly improved so that air-cooled engines are now running at very much better economies than their competing water-cooled engines. Not only that, but they have proved themselves capable of running supercharged at sea level to brake mean effective pressures which water-cooled engines have not been able to withstand, and they can also be supercharged at altitude without any added complications. Air-cooled-engine design is now advanced to the point where cylinders are superior in cooling characteristics to water-cooled cylinders. In consideration of the design of aircraft engines engineers started out basically with the thought that, in an airplane, we start off with a definite quantity of heat in the fuel-tank,

which, at the end of a flight, has been dissipated. That which went into useful work through the propeller was dissipated to the atmosphere directly. That which was lost through the exhaust-stacks as heat rejected because of the cycle employed is likewise dissipated directly to the atmosphere. That which was transferred through the cylinders and pistons was also dissipated to the atmosphere, directly in the air-cooled engine and indirectly in the water-cooled engine.

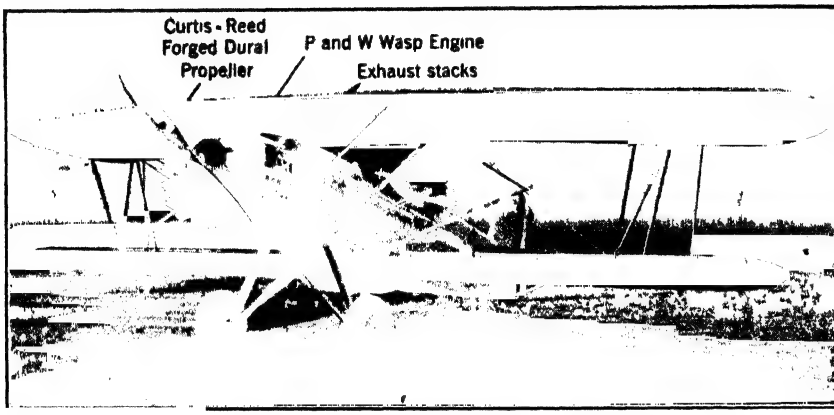


Fig. 230.—Diagram Showing Installation of the Pratt & Whitney "Wasp" Engine in Curtiss Falcon Airplane. Note Method of Cowling Engine and Exposure of Flanged Cylinders to Propeller Slipstream to Secure Air Cooling.

Air-Cooling Efficiency.—The proportion of the heat rejected through the exhaust and through the cylinder-walls and pistons can be varied within narrow limits. Fundamentally, the design of an aircraft engine is a matter of heat-flow. Our inability to get rid of this heat easily has resulted in the development of special materials for valves, pistons, piston-rings, cylinder-heads, valve-seats, valve-guides, valve-springs, and all the other parts of the engine. It was felt that if engineers could once solve the problem of heat-flow and get this rejected heat out to the atmosphere properly we would simplify the resulting mechanical problems greatly. Designers therefore took very great pains in the design of air-cooled cylinders, making provison for every possible means of cooling as the writer will discuss more in detail later. We began to regulate the oil-supply to the pistons and to design these pistons to permit a ready transfer of heat to the lubricating-oil below the point where an oil-cooler would be required. Care was taken not to carry this beyond that point. The final result was very satisfactory. In the new air-cooled-cylinder design the cooling is so successful that the problems of overheated valves, burned pistons, stuck rings, and burned cylinder-heads have been simplified greatly. This is one of the outstanding achievements of air-cooled engine development because it not only permits high power-output per cubic inch of displacement but also simplifies to a great degree the mechanical problems.

Radial Cylinder Placing Ideal for Air Cooling.—Aircraft engines especially, present important inducements for air cooling, in conjunction with a fixed radial or rotary type of engine. In such designs the cylinders are

each exposed to substantially the same amount of air-flow as shown at Fig 230 and the air velocities in the propeller slipstream attain very high values. Furthermore, the load on the engine decreases with a decrease in speed, so that it is possible to operate aircraft air-cooled radial or rotary engines at mean effective pressure as high as can be done with water-cooled engines. There is no doubt, therefore, that air cooling of aircraft engines of the radial or even the in-line or Vee types will continue to find favor with designers of such engines.

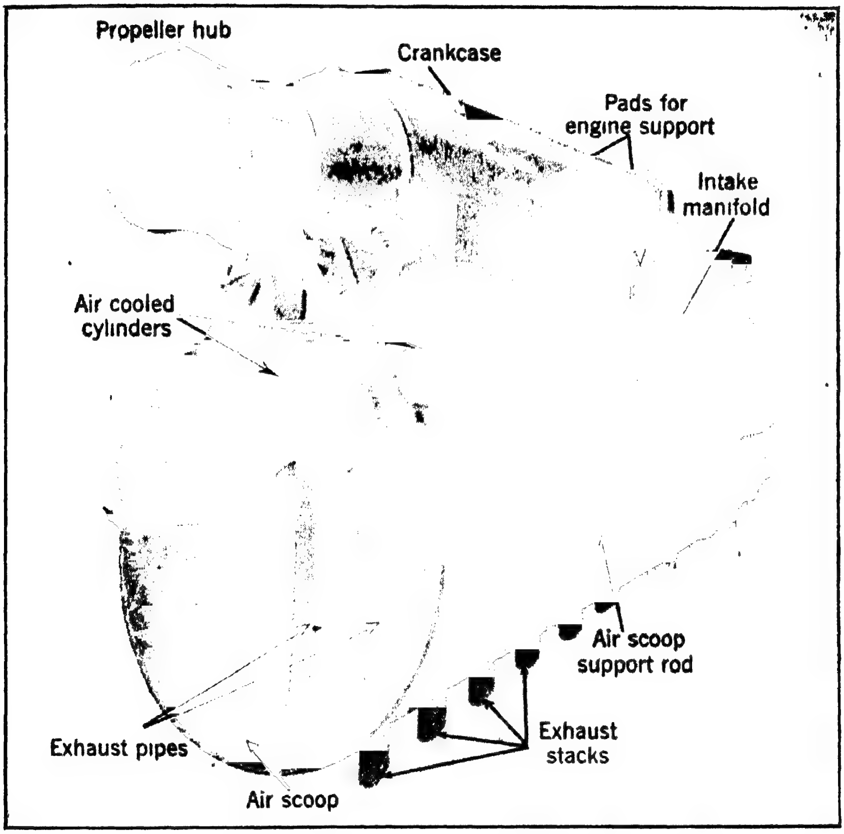


Fig. 231.—Wright "Vee" 1456 Air-Cooled Engine is of the Inverted Type and has Twelve Air-Cooled Cylinders. Note Use of Cowling to Direct Air Blast from the Propeller to the Space between the Cylinders and Against the Heated Exhaust Pipes.

It is stated that a representative medium-size six-cylinder automobile engine is provided with a radiator, the wetted surface of which totals something like 8,000 square inches; that is, the air drawn through the radiator comes in contact with that much surface which is available for transferring the heat from the water to the air. If this engine was made without any water jackets and cylinders left plain the external area of all the cylinders would total something like 400 square inches. In other words, the air would have to take away twenty times more heat from a given area of metal. In air-cooled engines it is therefore necessary to increase the effec-

tive cooling area of the cylinder walls greatly. This is done as a rule by forming a large number of cooling fins or flanges integral with the cylinder as shown clearly in Fig. 224 and using material for the cylinder head that is a very good conductor of heat and that is light, such as aluminum alloy.

In the air-cooled field we have not yet reached the ultimate type, but have at least put cylinder cooling and valve troubles behind us, and are now attacking the problems of master rod and articulated rod construction that are involved and that must be considered in the tendency toward higher speeds of revolution. The air-cooled piston is subjected to considerably higher temperatures than the piston of the water-cooled engine and, on account of the motion of the master rod, much greater angularity prevails in the connecting rods of a nine-cylinder engine, numbers four, five, six, seven. This angularity causes greater side pressure on these pistons and the master rod piston also is subjected to special side pressure, due to the action of the other cylinders upon the rod. All these conditions must be met by a balanced design, which will maintain the outside diameter within as small limits as possible and yet provide satisfactory conditions for the operation of the parts just mentioned. The big end of the master rod must be given sufficient stiffness to permit of higher rotative speeds than are used at present.

Vee Type Air-Cooled Engines.—The Vee type air-cooled engine as shown at Fig. 231 on the other hand, now that its cylinder and valve problems are solved, can follow the general lines of water-cooled technique and will, authorities believe, during the next few years become a very popular type wherever overall length is not the deciding factor.

One of the most interesting developments of recent years is the air-cooled Liberty, referred to earlier in this paper, and developed by Messrs. Jones and Heron. This is a most satisfactory powerplant, developing 420 horsepower at 1,800 r.p.m., but necessarily somewhat heavier than would be the case if it were a completely new design and not an adaption, but nevertheless 200 pounds lighter than the water-cooled Liberty with radiator and water. As 10,000 new Liberty engines are still in storage, and as the water jackets are beginning to corrode and become unreliable, the importance of this development from a commercial and economic viewpoint, as well as for its military value, is quite evident. In order to obtain sufficient space between the cylinders for proper air cooling, it was found necessary to reduce the cylinder bore to 4.625 inches instead of five inches as formerly, and to get the power by higher mean effective pressure and higher revolutions. With the water-cooled Liberty 1,700 r.p.m. was the highest speed at which the engine could run satisfactorily, but with the lighter reciprocating parts and smaller piston areas of the air-cooled engine, speeds of 1,800 r.p.m. are satisfactorily maintained. Barring a few minor changes, the rest of the engine, except the cylinders, valve gear, pistons and induction system, remains the same.

The cooling is obtained by means of a scoop, as in some of the early English and French air-cooled Vee type engines, which provides easy means of shuttering in cold weather. The distribution of cooling air to the

various cylinders is much less troublesome than was anticipated, no baffles or deflectors being necessary. The enclosed valve gear will be noted in all these engines, due to the belief that the operating parts should be properly lubricated and the valve springs protected from spray when used in seaplane installations. Failure of exhaust valve springs has often resulted from the sudden chilling due to spray from the floats or propeller. Another important reason for the enclosed valve gear is the ease with which an enclosed valve gear lends itself to compensation, a very important item if durability is a consideration. Wind tunnel experiments have shown that the gear on the top of an air-cooled cylinder offers considerable resistance and the gain in head resistance by enclosing these parts in streamlined boxes is by no means negligible.

With the Vee type, it is necessary to provide cowling to direct the air-flow to the cylinders; it is not necessary to use a blower for this purpose though the early Renault air-cooled engine previously described used a blower for the air blast. Engines of this type are not apt to be lighter per horsepower than the radial type even though counterweights are eliminated. The higher crankshaft-speed possible by the elimination of the master rod, and therefore the higher power, is counteracted by the added weight of the long crankcase and shaft required by the cylinder spacing. Because of the addition of cowling as well as the increased number of cylinders, the Vee type is not so accessible nor so readily dismantled as are radial engines, which are inherently simple from a maintenance standpoint. Heretofore, the drag of the cylinders was supposed to have been considerably greater than that of the radiator usually required for a water-cooled engine. This assumption has been disproved, as similar airplanes equipped with both air- and water-cooled engines have shown a considerable increase in speed in favor of the air-cooled type, as has previously been brought out. This increase is particularly noticeable at high altitudes when the ordinary radiator is shuttered. High speed is possible with the radial type of engine, as shown by the Gourdon airplane, which is reported to have reached a maximum speed of 224 m.p.h. with a 450 horsepower radial engine.

Limitations to Air Cooling Possible.—Engine-speed, some authorities claim, is the one factor that may give the water-cooled engine a chance to survive in aeronautics. Obviously an engine of given size, if run twice as fast and at the same mean effective pressure, will give nearly twice the power. Possibly the limiting mechanical features may not differ greatly between the two types, although when five to nine cylinders are working on one crankpin the limiting speed seemingly is bound to be lower than when two cylinders are working on one crankpin. However, granting the contention that this problem can be solved and that the air-cooled and water-cooled engines will be equal in this respect, the question of cooling remains, Can the air-cooled engine be cooled indefinitely as the speed increases? The cooling required on a given cylinder is a function of horsepower per unit of heat-dissipation area, and the direct air-cooled engine has a definitely limited dissipation area.

The diagram at Fig. 232 illustrates the fact that the useful speed of the air-cooled engine tested seems to have a definite limit and that this is

the speed of rotation, or about 1,900 r.p.m., where the mean effective pressure drops rapidly with a corresponding rise in fuel consumption. These curves, which were plotted as a result of a test with an engine having a cylinder displacement that gave a power output per cylinder of between 25 and 28 horsepower do not represent the last word, by any means and recent improvements in air-cooled engine designs show that the critical speed at which the drop occurs can be materially increased. It should be borne in mind that any engine cylinder, air- or water-cooled, will have a critical speed and this depends on other factors of design besides cooling, though heat dissipation is one of the most important considerations.

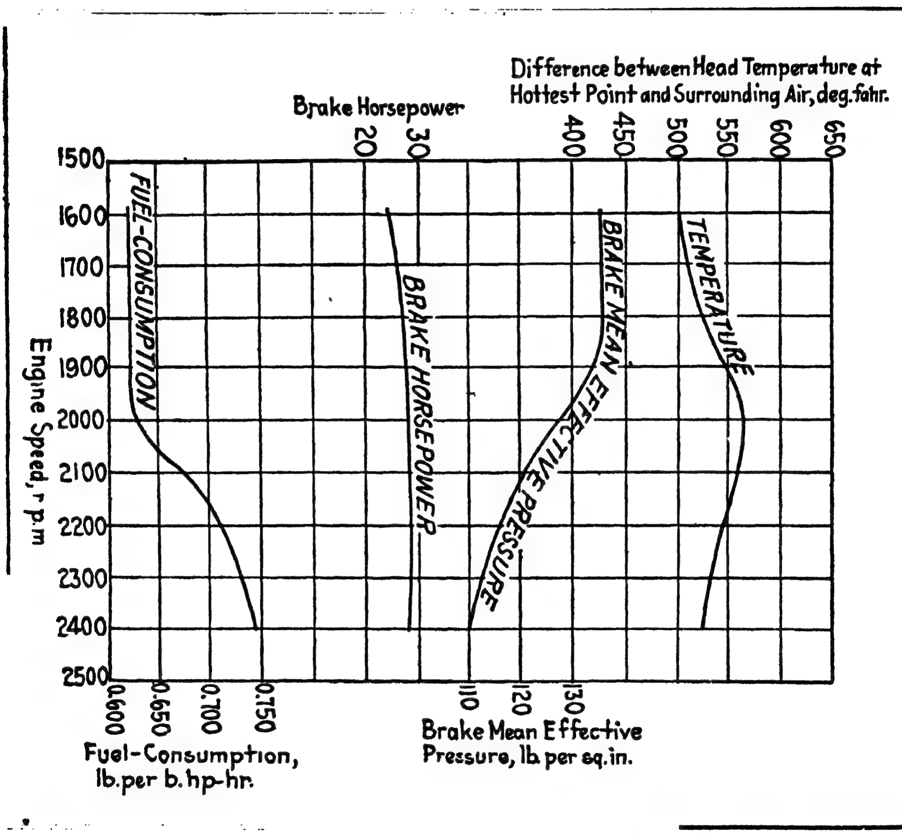


Fig. 232.—Performance Curves of an Air-Cooled Cylinder. These Curves Demonstrate that the Useful Speed of Such a Cylinder has a Definite Limit. This is the Speed Above which the Mean Effective Pressure Drops Rapidly with a Corresponding Rise in Fuel Consumption.

High Engine Speeds Favor Water Cooling.—The discussion on high crank-speed goes on continuously. The rise of the air-cooled engine forced the water-cooled engine to high crank-speeds so that it could compete, but, we must remember that the permissible crank-speed is a function also of the airplane design and air screw efficiency. J. G. Vincent said in a paper that we could count on about ten r.p.m. per m.p.h., perhaps, for the

best air screw, efficiency and fuel economy. That being the case, in a 125-m.p.h. airplane, 1,200 r.p.m. is probably the best propeller-speed. Yet for low weight per horsepower engineers are turning these engines at 2,000 r.p.m. The solution advanced by some is to gear the propeller down. That idea was brought out in Navy PN9 and PN10 airplanes in which, with the reduction gearing installed in Packard engines that are turning 2,500 r.p.m. in level flight, the planes are able to get off the water with tremendously heavy loads because of the high propeller-thrust for take-off and to get high propeller efficiency for long-range flights. The secret of the success of the PN9 was two-to-one reduction gearing.

When engineers undertake these different designs, however, they are more or less uncertain. Does the weight of the propeller and reduction gearing counter-balance the improvement in thrust? No definite figures are available to work from except that we have certain examples of improved performance with the use of reduction gearing. For that reason, when Pratt and Whitney engineers started to build the Hornet, they incorporated a two-to-one reduction gear but realized that the gear was useless unless conditions permitted turning the engine up to high speed. Again, the question arises whether the ratio should be two to one or five to three or some other. Small engines have used as high as three to one reduction ratio.

Designers thought they were doing rather well when they turned the water-cooled engines at 2,100 r.p.m. Then they increased the speed to 2,500, and now it is 2,800 r.p.m. One of the Packard engines developed 500 horsepower at 2,100 r.p.m., passed a 50-hour test, with wide-open throttle, with only two stops, developing 600 horsepower at 2,500 r.p.m., and the Packard racing engine recently showed 700 horsepower at 2,800 r.p.m. Engineers felt the same way about the radial engines. Several years ago it was believed that the model J was limited to 1,800 r.p.m., but now they are turning it up at 2,100 and 2,200 r.p.m. and diving the engine at 2,500 r.p.m. The engine-speed will be in relation to the level-flight propeller-speed. In the case of the Wright Whirlwind engine which turns at 1,900 r.p.m. in level flight; in a dive the highest speed recorded was about 2,400 r.p.m. in a straight-down dive for a good many thousand feet.

Also with regard to the slower revolutions of the radial engines, the reader should remember that the propellers will turn at slower and more efficient speed because they will sweep a larger area. More than that, the diameter of the engine with the larger propeller is of less retarding effect because the central part of the slow-speed propeller is not efficient and might as well be cowled-out. With the larger propeller and the slower speeds, the problem of the greater diameter of the radial engine is solved and some engineers claim that the gain from the propeller on direct drive will more than make up for the greater engine-speed of the water-cooled engine with the addition of gears to secure propeller speeds that will be efficient.

In the question of powerplant economy, we must consider also the propeller efficiency. It is well known that propeller design is largely dictated by the speed of advance of the aircraft. In airplanes making a high speed of 125 m.p.h., the propeller speed should not be greatly in excess of

1,250 r.p.m. without loss in economy. If now we want to employ the high powers resulting from high crankshaft-speeds, we must use reduction-gears. This was the basic idea that dictated the engine installation of the water-cooled West Coast-Hawaiian PN9's. With Packard 1A-1,500 engines turning between 2,200 and 2,400 r.p.m. in level flight, we attained very high propeller-efficiency which resulted in increased range. At the same time high thrust on take-off was obtained which permitted us to get off with a very large percentage of useful to full load. The final result was the world's seaplane endurance-record of 28 hours 35 minutes. In other words, we are thinking now in terms of "powerplant weight per thrust horsepower-hour." In fast airplanes it is permissible, of course, to turn

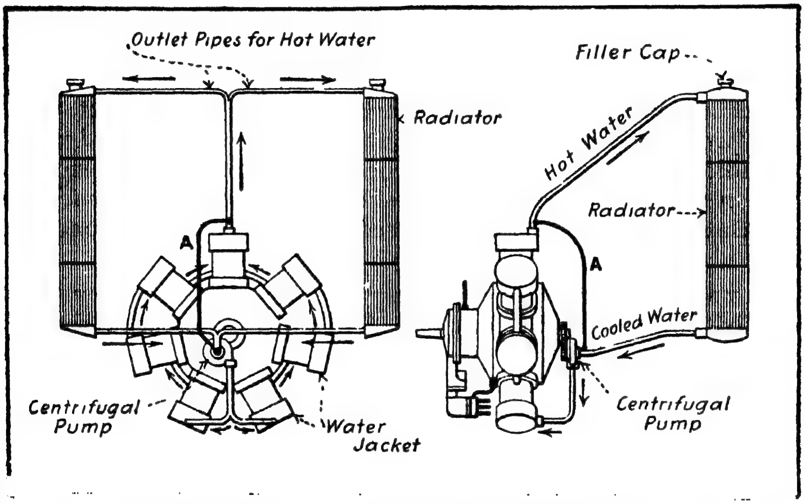


Fig. 233.—Diagram Showing How a Seven-Cylinder Radial Airplane Engine May be Water-Cooled. System Shown is that of Salmson Powerplant.

the engines at high speed without the use of gearing. Our pursuit airplanes are now being so operated, utilizing crankshaft-speeds as high as 2,250 r.p.m. successfully. Now, in a steep dive at full throttle, the engine turns up to very high crankshaft-speeds. Our pursuit engines are perfectly capable of withstanding these high speeds which may be as high as 3,500 r.p.m. during a steep dive.

Cooling Systems Generally Applied.—There are two general systems of engine cooling in common use, that in which water is heated by the absorption of heat from the engine and then cooled by air, and the other method in which the air is directed onto the cylinder and absorbs the heat directly instead of through the medium of water. When the liquid is employed in cooling it is circulated through jackets which surround the cylinder casting and the water may be kept in motion by two methods. The one generally favored is to use a positive circulating pump of some form which is driven by the engine to keep the water in motion. The other system seldom used in aviation engines is to utilize a natural principle that heated water is lighter than cold liquid and that it will tend to rise to the top of the cylinder when it becomes heated to the proper tempera-

ture and cooled water takes its place at the bottom of the water jacket. Air-cooling methods may be by radiation or convection. In the former case the effective outer surface of the cylinder is increased by the addition of flanges machined or cast thereon, and the air is depended on to rise from the cylinder as heated and be replaced by cooler air. This, of course, is found only on stationary engines. When a positive air draught is directed against the cylinder by means of the propeller slipstream in an airplane or the cylinders revolve around a stationary crankshaft as in the Gnome

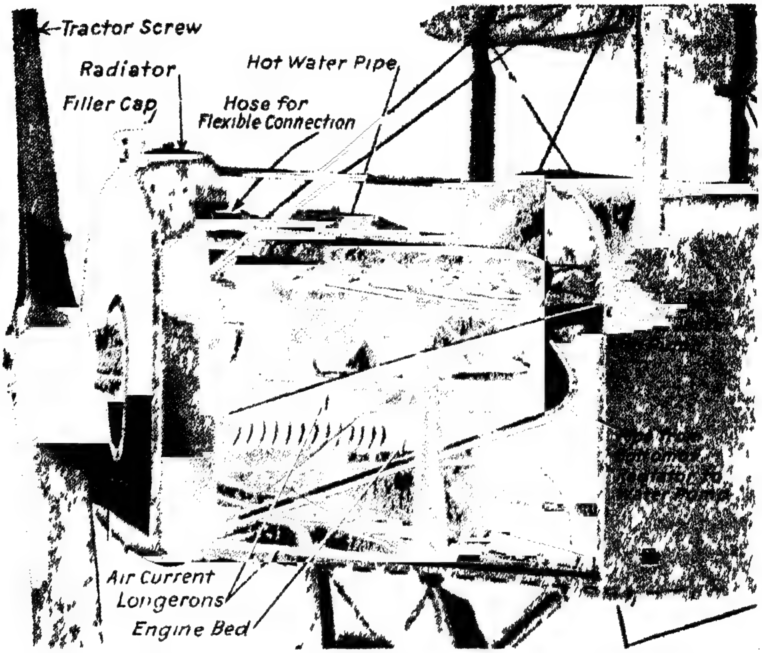


Fig. 234.—How Water-Cooling System of Early Airplane Engine is Installed in Fuselage, Following the Then-Current Automobile Practice. Radiator is Mounted in Front of the Engine and Directly Back of Propeller.

construction, cooling is by convection and radiation both. Sometimes the air draught may be directed against the cylinder-walls by some form of jacket which confines it to the heated portions of the cylinder as shown at Fig. 231, where a cowling confines and directs a large portion of the slipstream around the flanged cylinders and heads.

Cooling by Positive Water Circulation.—A typical water-cooling system in which a pump is depended upon to promote circulation of the cooling liquid is shown at Figs. 233 and 234. The radiator is carried at the front end of the fuselage in some cases, and serves as a combined water tank and cooler, but in other designs it is carried at the side of the engine, as in Fig. 235, or attached to the central portion of the aerofoil or winged structure or placed below the engine in the fuselage. In racing planes, the radiator may form part of the wing covering. Radiators are composed of an upper and lower portion joined together by a series of pipes which

may be round and provided with a series of fins to radiate the heat, or which may be flat in order to have the water pass through in thin sheets and cool it more easily. Cellular or honeycomb coolers are composed of a large number of bent or zig-zag tubes which will expose a large area of surface to the cooling influence of the air draught forced through the radiator either by the forward movement of the airplane or by some type of fan in automobiles and the propeller slipstream in aircraft. The cellular and flat tube types have almost entirely displaced the flange tube radiators which were formerly popular because they cool the water more effectively, and may be made lighter than the tubular radiator could be for engines of the same capacity, a very important advantage in airplanes.

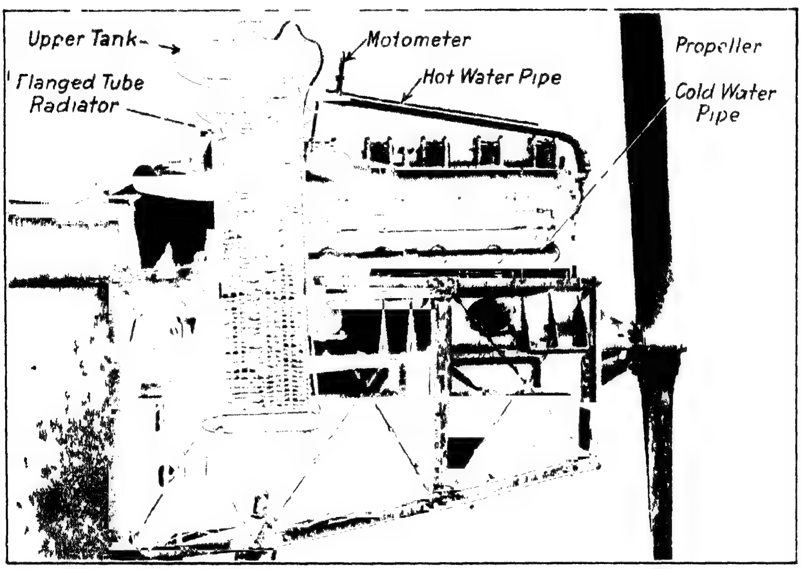


Fig. 235.—One of the Earliest Attempts to Reduce Resistance of Front Radiator. Narrow Finned Tube Radiator Installed at the Side of Early Hall-Scott Airplane Powerplant, Mounted in a Standard Fuselage.

In the simplest systems water is drawn from the lower header of the radiator by the pump and is forced through a manifold to the lower portion of the water jackets of the cylinder. It becomes heated as it passes around the cylinder walls and combustion-chambers and the hot water passes out of the top of the water jacket to the upper portion of the radiator. Here it is divided in thin streams and directed against comparatively cool metal which abstracts the heat from the water. As it becomes cooler it falls to the bottom of the radiator because its weight increases as the temperature becomes lower. By the time it reaches the lower tank of the radiator it has been cooled sufficiently so that it may be again passed around the cylinders of the motor.

The popular form of circulating pump is known as the "centrifugal type" because a rotary impeller of paddle-wheel form throws water which it receives at a central point toward the outside and thus causes it to

maintain a definite rate of circulation. Such a pump is clearly shown in section at Fig. 220 which also shows the method of driving it. The pump is always a separate appliance attached to the engine and driven by positive gearing or direct-shaft connection. The centrifugal pump is not as positive as the gear form, and some manufacturers prefer the latter because of the positive pumping features especially in oiling systems. They are very simple in form, consisting of a suitable cast body in which a pair of spur pinions having large teeth are carried. One of these gears is driven by suitable means, and as it turns the other member they maintain a flow of water around the pump body. Gear pumps of adequate capacity for

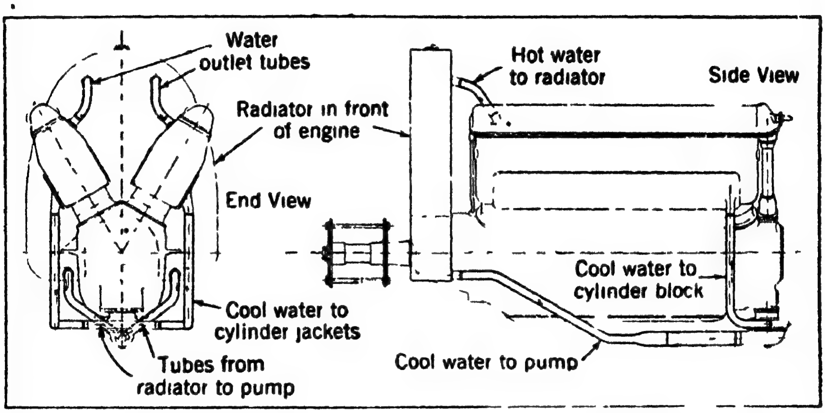


Fig. 236.—Water-Cooled System Showing Simple Piping When Block Castings are Used for Cylinders. Radiator Mounted in Front of the Engine.

cooling aviation engines would be heavier than a centrifugal pump so the latter are almost universally used for circulating cooling water though gear pumps are preferred for oil circulation. The pump should always be installed in series with the water pipe which conveys the cool liquid from the lower compartment of the radiator to the coolest portion of the water jacket.

Water Circulation by Natural System.—Some automotive engineers contend that the rapid water circulation obtained by using a pump may cool the cylinders too much, and that the temperature of the engine may be reduced so much that the efficiency will be lessened. For this reason there was a tendency to use the natural method of water circulation where the cooling liquid is supplied to the cylinder jackets just below the boiling point, and the water issues from the jacket at the top of the cylinder after it has absorbed sufficient heat to raise it just about to the boiling point.

As the water becomes heated by contact with the hot cylinder and combustion-chamber walls it rises to the top of the water jacket, flows to the cooler, where enough of the heat is absorbed to cause it to become sensibly greater in weight. As the water becomes cooler, it falls to the bottom of the radiator and it is again supplied to the water jacket. The circulation is entirely automatic and continues as long as there is a difference in temperature between the liquid in the water spaces of the engine

and that in the cooler. The circulation becomes brisker as the engine becomes hotter and thus the temperature of the cylinders is kept more nearly to a fixed point. With the thermosyphon system the cooling liquid is nearly always at its boiling point, whereas if the circulation is maintained by a pump the engine will become cooler at high speed and will heat up more at low speed.

With the thermosyphon, or natural system of cooling, more water must be carried than with the pump-maintained circulation methods. The water spaces around the cylinders should be larger, the inlet and discharge water manifolds should have greater capacity, and be free from sharp corners which might impede the flow. The radiator must also carry more water

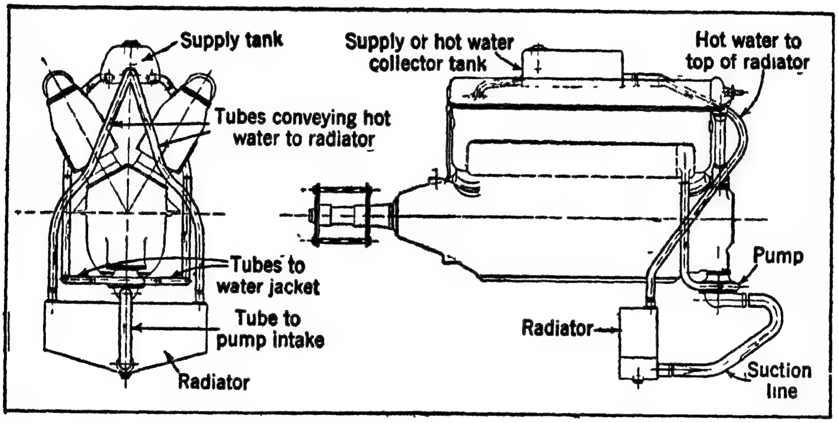


Fig. 237.—Diagrams Showing the Mounting of the Radiator Below the Engine and the Use of a Supplementary Supply Tank Placed Above the Motor Cylinders.

than the form used in connection with the pump because of the brisker pump circulation which maintains the engine temperature at a lower point. Consideration of the above will show why the pump system is almost universally used in connection with airplane powerplant cooling. It is also more popular in automobile applications and few modern cars are now cooled by the natural system.

Radiator Location.—Various locations of the water cooling radiators have been proposed. The method of piping the twelve-cylinder Vee engine of Hispano-Suiza design shown at Fig. 236 indicates that the use of a frontal radiator permits of the most simple and direct piping. When a cooling radiator is installed below the engine, as shown at Fig. 237, more piping is necessary because a feed tank must be carried above the water jackets to insure a gravity head to the water pump. The same system, modified in that the radiators are placed below the engine but at the side of the fuselage is shown at Fig. 238. The auxiliary tank in this system is not in series with the radiator as indicated in the piping arrangement shown at Fig. 237 but is connected in parallel or a shunt connection with the outlet pipes from the jackets. When used in this manner, the auxiliary tank acts as a steam condenser.

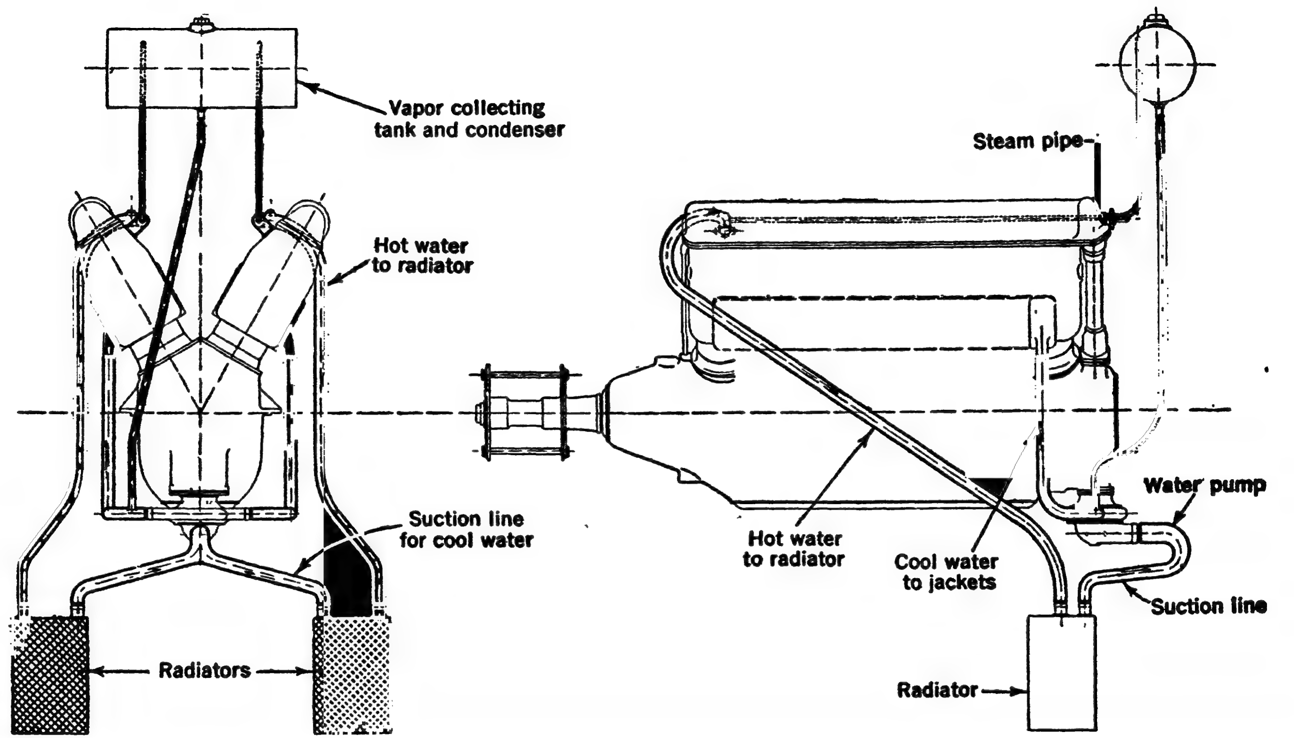


Fig. 238.—Installation Diagram Prepared by Hispano-Suiza Engineers, Showing Water-Cooling System in which Radiator is Mounted Below and a Vapor Collecting and Condensing Tank is Mounted Above the Engine. This System Permits of Running the Engine at Higher Temperatures than Possible with the Simpler System Because Arrangements are Made to Trap and Condense the Steam.

Regulating the water temperature may be done in a number of ways, depending on the radiator installation. When the radiator is mounted ahead of the engine, as in Fig. 239, the water-flow may be controlled by a manually controlled or thermostatically actuated valve introduced in the water outlet pipe as at A or the air-flow through the radiator interstices may be regulated by shutters as at B. When the radiator is placed below the engine and a special auxiliary feed tank is mounted above the motor we have the conditions shown diagrammatically at Fig. 240. In one series of installations the feed tank is in front, in the other it is in back of the

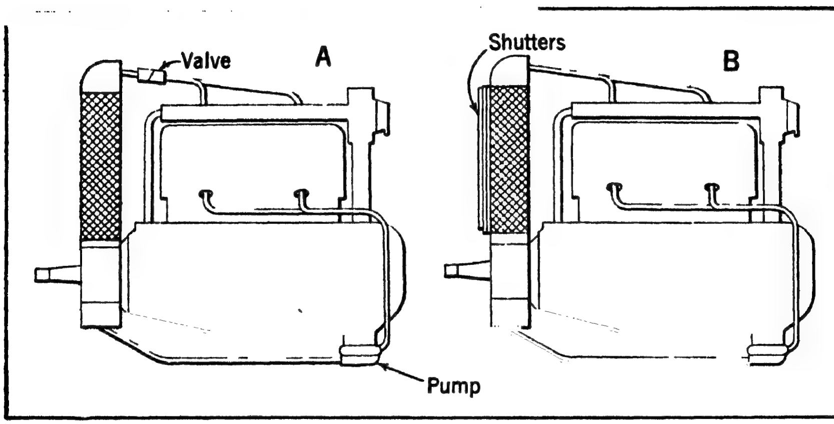


Fig. 239.—Diagram Showing Simple Methods of Regulating Water Temperature. A.—Shut-Off Valve in Warm Water Outlet Pipe. B.—Shutters Mounted in Front of the Radiator.

motor, otherwise the conditions are the same. At A, the shut-off valve is used in the pipe line running from top of water jacket to auxiliary feed tank. At B, the entire radiator may be raised into the fuselage or lowered at will. As it is raised, it is shielded from the air stream and the amount of exposed area can be proportioned to the amount of cooling effect desired. This system involves trouble and mechanical complication. The use of shutters, as shown at C makes possible manual or automatic control with a small amount of mechanism and is generally used. The use of shutters in a modern pursuit plane is shown at Fig. 241.

Resistance of Radiators.—The head resistance of the radiator can be assumed, for usual purposes, proportional to the square of the flying speed and to the density of the air. For unobstructed positions it may be represented by the equation

$$R = b\,p\,V^2 \text{ where}$$

R = the head resistance in pounds per square foot, or kilograms per square meter, of frontal area

V = the airplane speed in miles per hour, or in meters per second

p = the air density in pounds per cubic foot, or grams per cubic centimeter

b = a constant for each type of radiator.

The value of b ranges between very wide limits. If the English units are

used, for cellular radiators with straight-sided air tubes, it ranges from 0.0009 for cores with very large free area to 0.0023 for types with very small free area; for fin-and-tube radiators, unless very open, it exceeds 0.0020; and for irregular types with turbulence vanes it may run as high as 0.0026. The corresponding values for the metric units are respectively 1.4, 3.5, 3.1 and 4.0.

When the radiator is placed in the nose of the fuselage as in Fig. 234 the head resistance chargeable to the radiator, meaning the difference between the head resistance of the complete airplane and the resistance that it might have had if it could have been designed without a radiator, is in general very large and, for a given flying speed, the head resistance chargeable to the radiator increases with increase in air-flow. It is considerably greater for a radiator placed in the nose of the fuselage than when the nose of the fuselage is streamlined and a radiator of equivalent cooling capacity is placed in an unobstructed position as in Fig. 241. A radiator placed in the wing increases, in general, the resistance, or drag, of the wing, but the increase is not large and there may even be a decrease at certain angles of attack. The radiator, unless shuttered, will inevitably decrease the lifting power of the wing because no vacuum lifting effect is present above that section of the wing where the radiator is installed and very little positive lift is obtained because of air-flow through the radiator. This does not apply to those forms where the radiator forms part of the wing skin or covering but only to radiators having open insterstices for air-flow.

If the radiator is placed in the slipstream, the head resistance chargeable to it is probably approximately equal to what it would be if the propeller could be removed and the airplane driven by some other means at such a speed as would give the same speed of air relative to the radiator. If this is true, the assumption that the speed of the slipstream relative to the radiator is from twenty to 25 per cent greater than the flying speed would apply to head resistance as well as it does to air-flow. Since head resistance varies as the square of the speed, an increase of twenty per cent in speed results in an increase of 44 per cent in resistance. The effect of the swirl of the slipstream is similar to that of yawing the radiator, which is to increase the head resistance for unobstructed positions and for positions that would be unobstructed but for the propeller. For positions that would be obstructed without the propeller, as in the nose of the fuselage, the effect of the swirl of the slipstream is to make the air-flow somewhat less than it would be if there were no swirl and, since for such positions the head resistance chargeable to the radiator increases with increased air-flow, the effect of the swirl probably compensates to a slight extent for the effect of the increased speed of air due to the propeller.

In the modern designs, when radiators are installed in front of the engine as in the Macchi M7 shown at Fig. 242 or the Curtiss Robin shown at Fig. 243, the shell is rounded to give a better streamline effect. The method shown at Fig. 244, in which the water radiators are used as wing covering at the center section and each side is the most efficient. When skin radiators are used, the resistance of the water cooling area is not much more than that of the wing covering normally employed and none of the lift is sacrificed.

In computing resistance of water-cooling system, we must take into consideration further the drag-area of the usual exposed radiator used for the water-cooled engine. This factor is considerable. We must even go farther and consider the drag resulting from the additional wing-area required to support the cooling-system of a water-cooled engine. When we take all these factors into consideration we find that the air-cooled engine compares not unfavorably with the water-cooled engine, and we have not

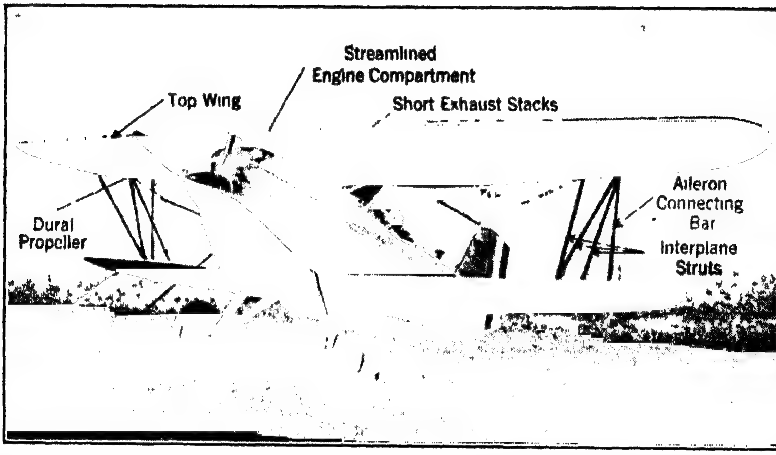


Fig. 240.—Modern Plane of Curtiss Design Showing How Carefully Motor is Streamlined in Nose of Fuselage. Note the Location of Radiator Provided with Shutters Streamlined Into the Lower Part of the Engine Compartment.

yet started to develop means of streamlining air-cooled cylinder heads which is possible as racing practice has shown in which cylinder heads have been cowled in by helmets of streamline form. A good example of cowling for cylinder heads of a radial engine is shown at Fig. 245 which shows a Bristol (English) single seat fighter with a short cowl in front of each cylinder head.

The radiator causes absorption of power in two ways, in carrying its weight and in overcoming its head resistance. It also has other adverse effects on the plane, among which are (*a*) obstruction of the pilot's view, (*b*) modifications in internal construction of the fuselage to accommodate the mounting of the radiator, and (*c*), in military machines, liability to injury from hostile fire. The power absorbed is composed of two parts: that due to head resistance and that due to weight. Since the head resistance varies approximately as the square of the flying speed, and the power can be measured by the product of a force and a velocity, the power absorbed due to head resistance varies approximately as the cube of the speed. Since, however, the weight is sustained by a "lift" on the wings, which is a constant and equal to the weight, and since this is accompanied by a "drag" that is proportional to the lift for a given angle of attack, the power absorbed due to weight is more nearly proportional to the first power of the speed, and is dependent upon the lift/drag ratio of the airplane. When

wing radiators are employed we find that parasitic resistance is reduced to the minimum because there is practically no head resistance and the fuselage has a much better entry than when a radiator is mounted above, in front of or below the engine. Radiators such as shown at Fig. 244 would be extremely vulnerable to enemy fire if used on military airplanes and also would be unfavorably affected by vibration and shocks due to landing on commercial airplanes.

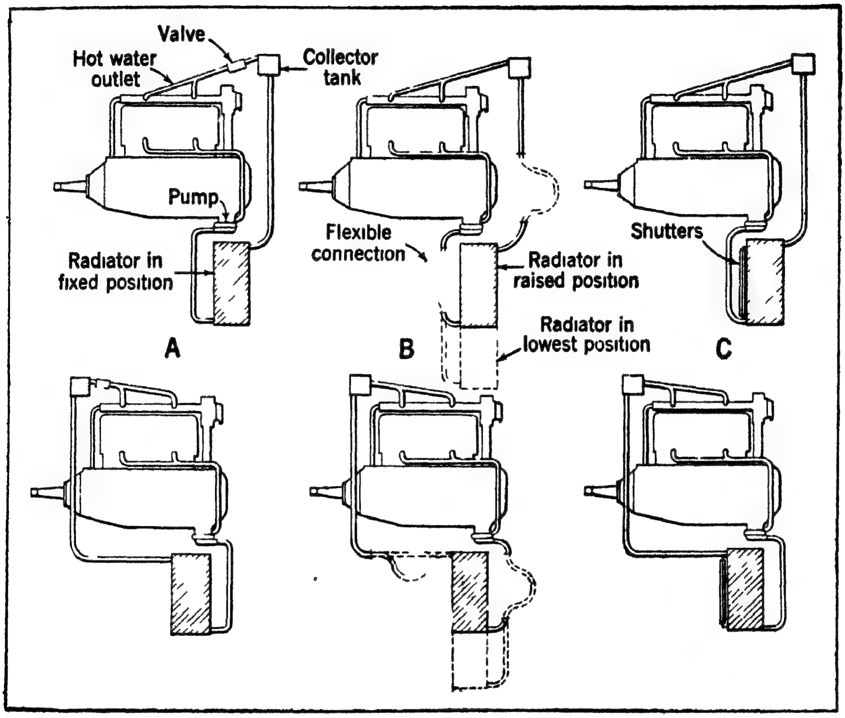


Fig. 241.—Diagram Showing Methods of Regulating Water Temperature in Water-Cooled Aviation Engines when Radiator is Mounted Below the Crankcase. A—Shut-off Valve in Hot Water Outlet Pipe. B—Movable Radiator which May be Raised from and Lowered into the Air Stream. C—Shutters Restricting Air Flow Through Radiator.

Direct Air-Cooling Methods.—The earliest known method of cooling the cylinder of gas-engines was by means of a current of air passed through a jacket which confined it close to the cylinder-walls and was used by Daimler on his first gas-engine. The gasoline-engine of that time was not as efficient as the later form, and other conditions which materialized made it desirable to cool the engine by water. Even as automobile engines have become more and more perfected there has existed an unreasonable prejudice against air cooling, though many forms of engines have been used, both in automobile and aircraft applications where the air-cooling method has proven to be very practical. The simplest system of air cooling is that in which the cylinders are provided with a series of flanges which increase the

effective radiating surface of the cylinder and directing an air current against the flanges to absorb the heat. This increase in the available radiating surface of an air-cooled cylinder is necessary because air does not absorb heat as readily as water and therefore more surface must be provided that the excess heat be absorbed sufficiently fast to prevent distortion of the

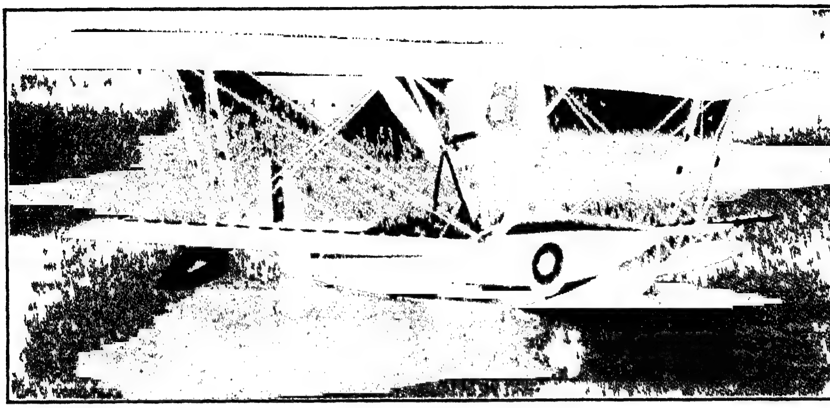


Fig. 242.—The Macchi M7 Seaplane with Engine Mounted Below Top Wing and Having an Inclined Radiator, Part of which is Covered by Movable Shutters for Temperature Regulation.

cylinders. Air-cooling systems are based on a law formulated by Newton, which is: "The rate for cooling for a body in a uniform current of air is directly proportional to the speed of the air current and the amount of radiating surface exposed to the cooling effect."

Air-Cooled Engine Design Considerations.—There are certain considerations which must be taken into account in designing an air-cooled engine, which are sometimes overlooked in those forms cooled by water.

Fig. 243.—The Curtiss "Robin" Cabin Type Monoplane has an OX5 Engine, with a Radiator Placed between the Engine and the Propeller. Note Rounding of Radiator Shell to Reduce Air Resistance.

Large valves must be provided to insure rapid expulsion of the flaming exhaust gas and also to admit promptly the fresh cool mixture from the carburetor. The valves of air-cooled engines are invariably placed in the cylinder head, in order to eliminate any pockets or sharp passages which would impede the flow of gas or retain some of the products of combustion and their heat. When high power is desired multiple-cylinder engines should

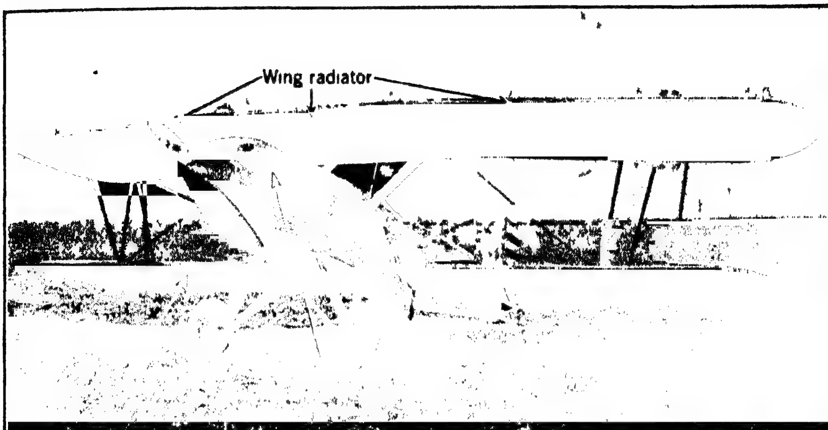


Fig. 244.—Curtiss XO13A Falcon Equipped with High Compression Curtiss "Conqueror" Engine and Wing Radiators for Cooling. This System Provides the Least Possible Parasitic Resistance Due to Cooling Water Radiators.

be used, as there is a certain limit to the size of a successful air-cooled cylinder. Much better results are secured from those having small cubical contents because the heat from small quantities of gas will be more quickly carried off than from greater amounts. All successful engines of the aviation type which have been air-cooled have been of the multiple-cylinder type and not to exceed six inches bore, though cylinders as large as eight inches bore have been cooled successfully.

An air-cooled engine must be placed in the fuselage, in such a way that there will be a positive circulation of air around the cylinder heads and a portion of the cylinders all the time that it is in operation. The air current may be produced by the tractor screw at the front end of the motor, or by a blower fan attached to the crankshaft as in the early Renault engine or by rotating the cylinders as in the Le Rhone and Gnome motors. Greater care is required in lubrication of the rotary air-cooled cylinders and only the best quality of castor oil should be used to insure satisfactory oiling as previously described.

The combustion-chambers must be proportioned so that distribution of metal is as uniform as possible in order to prevent uneven expansion during increase in temperature and uneven contraction when the cylinder is cooled. There must also be sufficient metal in the head to make for gradual heating and cooling and every part should be as liberally flanged as possible. It is essential that the inside walls of the combustion-chamber be smooth because any sharp angle or projection may absorb sufficient heat to remain incandescent and cause trouble by igniting the mixture before

the proper time. The best grades of cast nickel-iron or forged steel should be used in the cylinder and the machine work must be done very accurately so the piston will operate with minimum friction in the cylinder. The cylinder bore should not exceed eight inches and the compression pressure should not exceed 125 pounds absolute, or serious overheating may result unless design and fuel problems are very carefully worked out.

As an example of the care taken in disposing of the exhaust gases in order to obtain practical air cooling in early engines some cylinders were provided with a series of auxiliary exhaust ports uncovered by the piston when it reached the end of its power stroke. The auxiliary exhaust ports opened just as soon as the full force of the explosion was spent and a portion of the flaming gases was discharged through the ports in the bottom

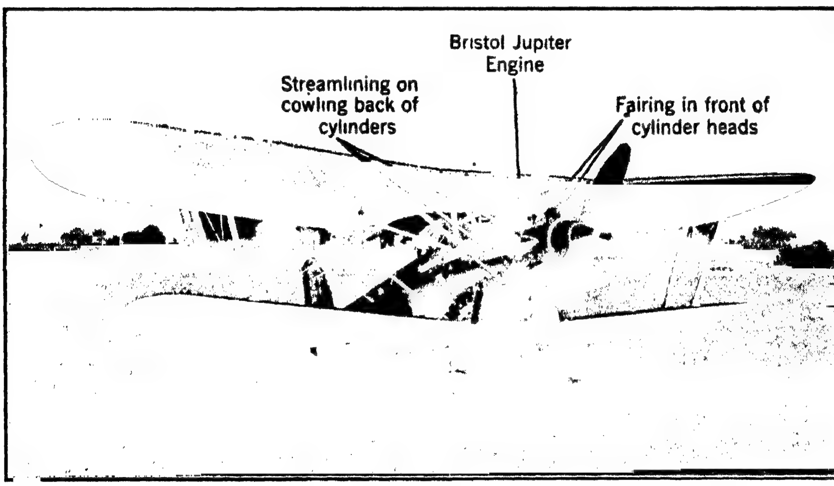


Fig. 245.—The Bristol Single-Seater Fighter Provided with Jupiter Air-Cooled Engine, Showing Method of Cowling and Streamlining Cylinder Heads to Reduce Parasitic Resistance.

of the cylinder direct to the air. Less of the exhaust gases remained to be discharged through the regular exhaust member in the cylinder head and this did not heat the walls of the cylinder nearly as much as the larger quantity of hot gas would. That the auxiliary exhaust port was of considerable value in early engines was conceded by many designers of fixed and fan-shaped air-cooled motors for airplanes, though modern engines do not require any such aid as an auxiliary exhaust to secure positive cooling. Salt and mercury cooled exhaust valve stems, alloy steel valve heads and properly cooled and well flanged cylinders and alloy heads have solved the problem.

Among the advantages stated for direct air cooling, the greatest is the elimination of cooling water and its cooling auxiliaries, which is a factor of some moment, as it permits considerable reduction in horsepower-weight ratio of the engine, something very much to be desired. In the temperate zone, where the majority of airplanes are used, the weather conditions change in a very short time from the warm summer to the extreme cold winter, and when water-cooled systems are employed it is necessary to

add some chemical substance to the water to prevent it from freezing. The substances commonly employed are glycerine, wood alcohol, or a saturated solution of calcium chloride. Alcohol has the disadvantage in that it vaporizes readily and must be often renewed. Glycerine affects the rubber hose, while the calcium chloride solution crystallizes and deposits salt in the radiator and water pipes. A new material, known as Prestone has been developed and used successfully for winter cooling by the U. S. Army and Navy, as well as some commercial operators using water-cooled engines.

BOILING POINT OF WATER AT VARIOUS ALTITUDES

Altitude Above Sea Level in Feet	Boiling Point in Degrees Fahrenheit
0.000	212
1,025	210
2,063	208
3,115	206
4,169	204
5,225	202
6,304	200
7,381	198
8,431	196
9,579	194
10,685	192
11,800	190

Air Cooling Permits Important Weight Saving.—All authorities agree that saving weight is the most important characteristic of air-cooled engines for aviation. Air-cooled engines save directly the weight of radiator, piping, water, pumps, shutters and radiator supports. Furthermore, there is saved in the plane structure itself the weight necessary to carry this water radiation equipment. This last saving is very important, as it runs from 50 per cent to 100 per cent of the direct saving. For instance, compare the weights and performances of air-cooled vs. water-cooled, two-seater planes built by the Chance Vought Corporation, both powered with Wright 200 horsepower engines. The air-cooled plane saves 145 pounds on radiator, shutters, piping and water and over 70 pounds more in plane structure. This weight saving has been applied to give the air-cooled plane higher speed and greater radius, without loss of ceiling or increase of landing speed. The air-cooled engine has increased the speed eighteen m.p.h. or fifteen per cent, and increased the cruising radius 150 miles or 51 per cent, notwithstanding the air-cooled plane carries 89 pounds more equipment than the water-cooled plane.

Air cooling is as logical for airplanes as water is for marine engines. Its simplicity, on account of the direct cooling, results in added dependability. This is the single most important reason for air cooling. In addition, a considerable saving of weight is effected that amounts to approximately 0.5 to 0.7 pounds per horsepower in the total powerplant weight. In no other way, at present, can this amount of weight be removed from the powerplant and at the same time increase its dependability. The reduction in powerplant weight is reflected in the airplane, both in the

structure of the fuselage and in the wing surface for a given loading. The reduction in the gross weight of the airplane is of the utmost importance for commercial operation, as it provides for carrying more pay-load with a given powerplant. From a military standpoint, this saving results in increased airplane performance. In addition to increased dependability and reduced weight, the radial type of air-cooled engine makes possible an aerodynamically superior and symmetrical fuselage, and gives a high center of thrust that allows ample propeller-diameter.

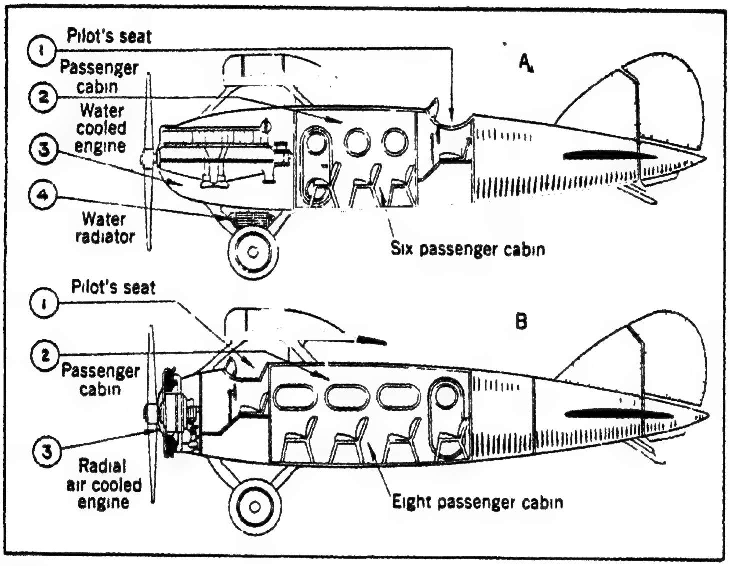


Fig. 245A.—Diagrams Showing Important Space Saving Possible With Air-Cooled Radial Engine.

Another fact regarding water cooling should be mentioned. William B. Stout stated that his associates had been driving airplanes on the Ford air routes with Liberty-twelve engines for about 1½ years. The ships made two round trips per day from Detroit to Cleveland and one round trip to Chicago. They have, as good mechanics as any in the world and Mr. Stout believed they knew how to mount a radiator and a Liberty engine; yet 90 per cent of all forced landings due to mechanical trouble have been caused by the plumbing and water troubles. Therefore, from actual operating experience with water cooling they are much in favor of the simpler air-cooled system and, while we may get greater power with water cooling, Mr. Stout is of the opinion that air cooling is an inevitable development for aircraft.

The planes used by the Ford-Stout airlines are now of the tri-motored all metal variety developed by Mr. Stout using air-cooled engines exclusively.

Comparing Radial and In-Line Engine Installation.—The illustrations at Fig. 245 A are diagrams adapted from some prepared and published by the makers of the Bristol Jupiter engine in order to show how powerplant is

simplified and space required for the engine is greatly reduced when a static radial engine is installed in a plane as compared to a similar plane with a motor in which the cylinders are in Vee or W arrangement. The machine shown is a Spad Berline. Comparing point one in each machine, we find that with a static radial engine as at B, two pilots have an excellent field of vision in a well placed cockpit, as compared with the seating arrangement shown at A in which the pilots are carried back of the cabin. The second point established is that the static radial engine allows the provision of seats for eight passengers as compared with the six passenger capacity in the plane with the longer in-line engine, which inevitably encroaches on the cabin space. Point three is made that ordinary inspection of the valve gear, etc., of a static radial engine may be made without removing the cowling. With a water-cooled or even an air-cooled Vee engine, removal of cowling is necessary in case of the former and some form of air-scoop in case of the latter. Then as a final point number four, attention is directed to the absence of a radiator shown in design A in the design outlined at B. Elimination of the radiator, water piping, pump, feed tanks, etc., certainly does simplify the engine installation.

Experience of U. S. Navy with Air Cooling.—One of the authorities on the use of air-cooled engines in the U. S. Navy, Commander E. E. Wilson gives some pertinent facts on air-cooled aviation engines which should be of general interest, especially as they are from an unprejudiced source, having at the time of publication no commercial interest in the promotion of such engines. Commander Wilson stated that in the development of naval aircraft, the weight per horsepower of the aircraft engine is manifestly of the greatest importance. This weight is important not only because of itself but also because, in any airplane, the weight of the powerplant is reflected back into the weight of the structure. The heavier the engine is for a given power, the greater the wing-area necessary to support it becomes and the greater the weight of the structure to withstand the loads. Carrying this thought a little farther, we find that the fuel-economy of an aircraft engine is of great importance. The lower the specific fuel-consumption is, the less is the quantity of fuel required for a given range and the less is the tankage capacity required to contain the fuel. This reduction in fuel quantity and in tank weight is likewise reflected back into the structure.

A brief comparison of powerplant weights between water-cooled and air-cooled engines shows that the water-cooled powerplant is heavier than the air-cooled powerplant by at least the weight of the cooling-system. This averages roughly 0.6 pounds per horsepower. The elimination of this cooling-system constituted the first point of attack after we had reduced the weights of the bare engines to what looked like the minimum consistent with reasonable dependability and durability. Again, since a careful analysis of engine failures indicates that a high percentage of the trouble is due to faults of the plumbing system, that is, to the fuel, the oil and the water pipe-lines, it is obvious that the elimination of one of these pipe-lines results in the elimination of about one-third of our troubles. The air-cooled engine is attractive, then, not only from its low powerplant-

weight per horsepower but also from its complete elimination of the troublesome water-cooling system.

Air-Cooled Engines in Pursuit Planes.—To demonstrate the possibilities of air-cooled engines in fighters, the Bureau of Aeronautics of the U. S Navy has been conducting airplane tests with air-cooled engines installed in high-speed aircraft. The Wright Apache airplane, a single-seat fighter equipped with the Pratt & Whitney Wasp engine, is a good example of what can be expected of an air-cooled engine installed in an airplane especially designed to accommodate it. This airplane shows speeds at least equal to those of the water-cooled fighters and a climb and a ceiling which are definitely superior.

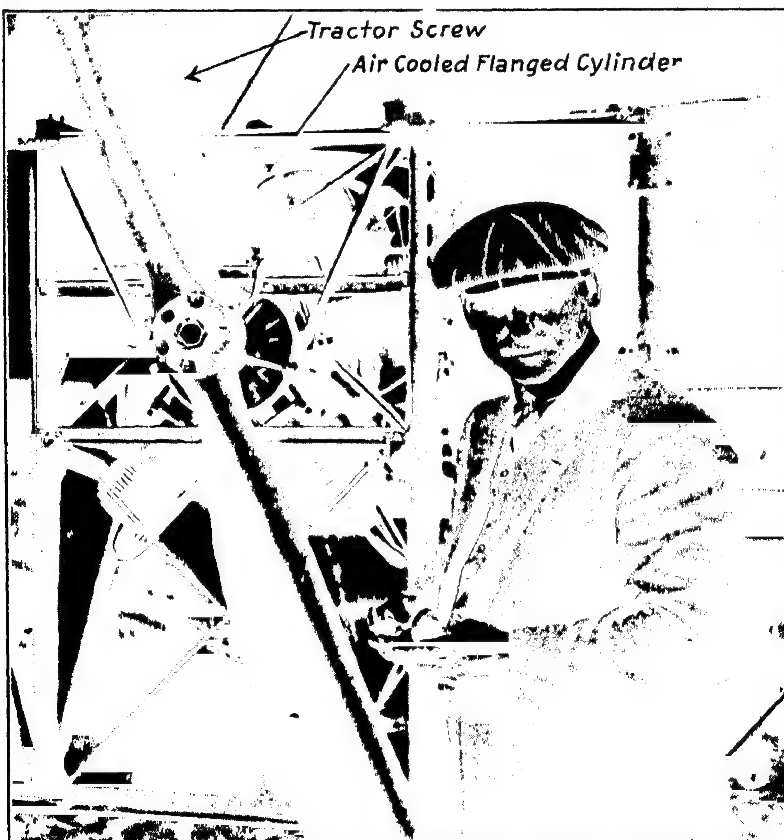


Fig. 246.—Pioneer Air-Cooled Engine Designer, Mr. Anzani of Paris, Testing an Early Five-Cylinder Air-Cooled Radial Aviation Motor, Installed in a Bleriot Monoplane. Note Exposure of Complete Engine to Propeller Slipstream and High Parasitic Resistance of Engine Mounting and Front End of Fuselage in Early Forms.

As a further test, Pratt & Whitney Wasp engines have been installed in Curtiss and in Boeing fighters designed originally for the Curtiss D12 water-cooled engine. In such makeshift installations we must accept some handicaps. Installation of the Wasp engine in the Curtiss fighter results

in a reduction in weight of 240 pounds, which corresponds exactly with the weight of the cooling-system of the original airplane; however, the engine must be installed farther ahead to maintain the balance, and this results in a slight increase in mounting weight. The reduction in powerplant weight reduces the landing-speed of the air-cooled fighter. If now we correct air-cooled fighter performance up to the landing-speed of the water-cooled airplane, we find their high speeds about equal. When we take these different installations and reduce them all to the same common denominator, we find that the air-cooled fighter is almost exactly equal to the water-cooled machine of equal power and that it attains this performance on a ten per cent reduction in gross weight.

In other words, the air-cooled engine has entered the pursuit field definitely to challenge the water-cooled engine, and this brings to mind the fact that our interest in racing has focused attention upon high speed at sea level as one of the important characteristics of an airplane. Manifestly, what we are interested in, particularly as pertains to fighters, is performance at the fighting altitude rather than at sea level, and here the air-cooled engine should have a decided advantage. The later types incorporate a rotary distribution-system which gives some supercharging at sea level and tends to maintain the mean effective pressure with increase in crankshaft speed. Performance at altitude is the proper criterion in comparing fighting aircraft. The Bureau of Aeronautics acts as a liaison between the operating squadrons of the fleet and the builders of the engines. It encourages criticisms on the part of the fleet of the machinery furnished. It has obtained intelligent recommendations from the fleet and has transferred them to the engine builder. As a result of this highly co-operative effort between the operating squadrons, the Bureau of Aeronautics and the aircraft-engine builders, we have today both air- and water-cooled engines which are cheap when measured by any or all of the requirements for aircraft engines. These requirements are (*a*) low weight per horsepower, (*b*) high economy in fuel, (*c*) maximum dependability, (*d*) maximum durability, (*e*) maximum ease in maintenance, (*f*) minimum cost, and (*g*) easy adaptability to quantity production.

Commander Wilson States.—"We are accustomed to think of air-cooled engines as radial engines and of water-cooled engines as in-line engines. There are such things, however, as water-cooled radial-engines and air-cooled in-line engines. In Navy development we have preferred the radial to the in-line air-cooled engine for a number of reasons. The short compact radial engine makes possible superior visibility for deck landings and bomb dropping, which is a fundamental requirement for naval aircraft. By virtue of its compactness, the radial engine has superior weight-per-horsepower characteristics, other things being equal. The in-line air-cooled engine may permit higher crankshaft-speed but, for our particular purposes, the radial engine seems more suitable. To date, the argument between the radial engine and the in-line air-cooled engine is purely academic and it will not be solved until in-line air-cooled engines have been developed and an actual comparison made. Basically, it is important to remember that compactness is a fundamental requirement for low weight. In general, a sphere

constitutes the optimum form from this viewpoint. Our radial engines, even when fitted with superchargers, approach this sphere in general form, while the in-line engines approach the square prism. In sizes up to at least 600 horsepower we prefer the radial engine.

"It will be seen that, insofar as air-cooled engines are concerned, we are rapidly obtaining a variety of air-cooled engines which will meet most of our requirements. The large air-cooled engines are now equipped with two to one reduction-gears and can be installed with geared or with direct drive in twin-engine patrol-airplanes. In developing the line of Pratt & Whitney engines the Naval Bureau of Aeronautics has kept in mind the desirability of interchangeability of parts between the engine types. In the two sizes of Pratt & Whitney engines, the two rear-ends are interchangeable. The advantages of this feature from the viewpoint of maintenance and operation are evident. It is our desire to extend the plan even farther so that all three engines will be as nearly interchangeable as their size will permit."

QUESTIONS FOR REVIEW

1. Why are engine cooling systems necessary?
2. Why is it important to reduce back-pressure in engine exhaust system?
3. Are airplane engine mufflers practical?
4. What are the two commonly used methods of airplane engine cooling?
5. Outline some advantages of air cooling.
6. Give some of the good features of water cooling.
7. Outline two systems of water cooling.
8. Why is radiator type and placing an important point to consider?
9. What type of radiator offers the least resistance?
10. Why are radiator shutters used?
11. Why should air-cooled cylinders be exposed to airstream?
12. What important savings are obtained by using static radial air-cooled engines?

CHAPTER XIX

AIRCRAFT ENGINE CYLINDER CONSTRUCTION

Methods of Cylinder Construction—Block Castings—Cylinder Grouping Influence on Crankshaft Design—Combustion-Chamber Design—Water-Cooled Cylinder Development—Wet Sleeve Construction—Valve Location—T Head Cylinders—L Head Cylinders—I Head Cylinders—Concentric Valves—Valve Operation—Methods of Driving Camshaft—Valve Springs—Valve Spring Surge—Four Major Contentions—Packard Multiple Cluster Valve Springs—Springless Valves—Knight Sleeve Valve Motor—Valve Design and Construction—Gas Velocity Effect on Power—Four Valves Per Cylinder—Valve Gears—Valve Gear Enclosure—Packard Oil-Cooled Valves—Bore and Stroke Ratio—Meaning of Piston Speed—Aviation Engine Crankshaft Speeds—Crankshaft Vibration Limits Speed—Inertia Forces Increase With Speed—Bearings Heat at High Speeds—Offset Cylinders.

The improvements noted in the modern internal-combustion motors for aircraft have been due to many conditions. The continual experimenting by leading mechanical minds could have but one ultimate result. The parts of the engines have been lightened and strengthened, and greater power has been obtained without increasing piston displacement. A careful study has been made of the many conditions which make for efficient motor action, and that the main principles are well recognized by all engineers is well shown by the standardization of design noted in modern powerplants. There are many different methods of applying the same principle, and it will be the purpose of this chapter to define some of the ways in which the construction may be changed and still achieve the same results. The various components may exist in many different forms, and all have their advantages and disadvantages. That all methods described are practical is best evidenced by the large number of successful engines which use radically different designs.

Methods of Cylinder Construction.—One of the most important parts of the gasoline-engine and one that has material bearing upon its efficiency is the cylinder unit. The type of construction followed is influenced by a number of considerations, the most important being the method of cooling employed. The cylinders may be cast individually as when air-cooled or in pairs, and it is possible to make all cylinders a unit or block casting as is invariably done in automobile practice though it is varied in detail in aviation engine practice. Some typical methods of cylinder construction are shown in accompanying illustrations. The appearance of individual cylinder castings of water-cooled form may be ascertained by examination of the early Hall-Scott airplane engine. Air-cooled engine cylinders are always of the individual pattern.

Considered from a purely theoretical point of view, the individual water-cooled cylinder casting has much in its favor. It is advanced that more uniform cooling is possible than where the cylinders are cast either in pairs or three or four in one casting. More uniform cooling insures that the expansion or change of form due to heating will be more equal. This is an important condition because the cylinder bore must remain true under

all conditions of operation. If the heating effect is not uniform, which condition is liable to obtain if metal is not evenly distributed, the cylinder may become distorted by heat and the bore be out of truth. When separate cast cylinders are used with integral water jackets it is possible to make a uniform water space and have the cooling liquid evenly distributed around the cylinder. In multiple-cylinder castings this is not always the rule, as in many instances, especially in four- and six-cylinder block motors where compactness is the main feature, there is but little space between the cylin-

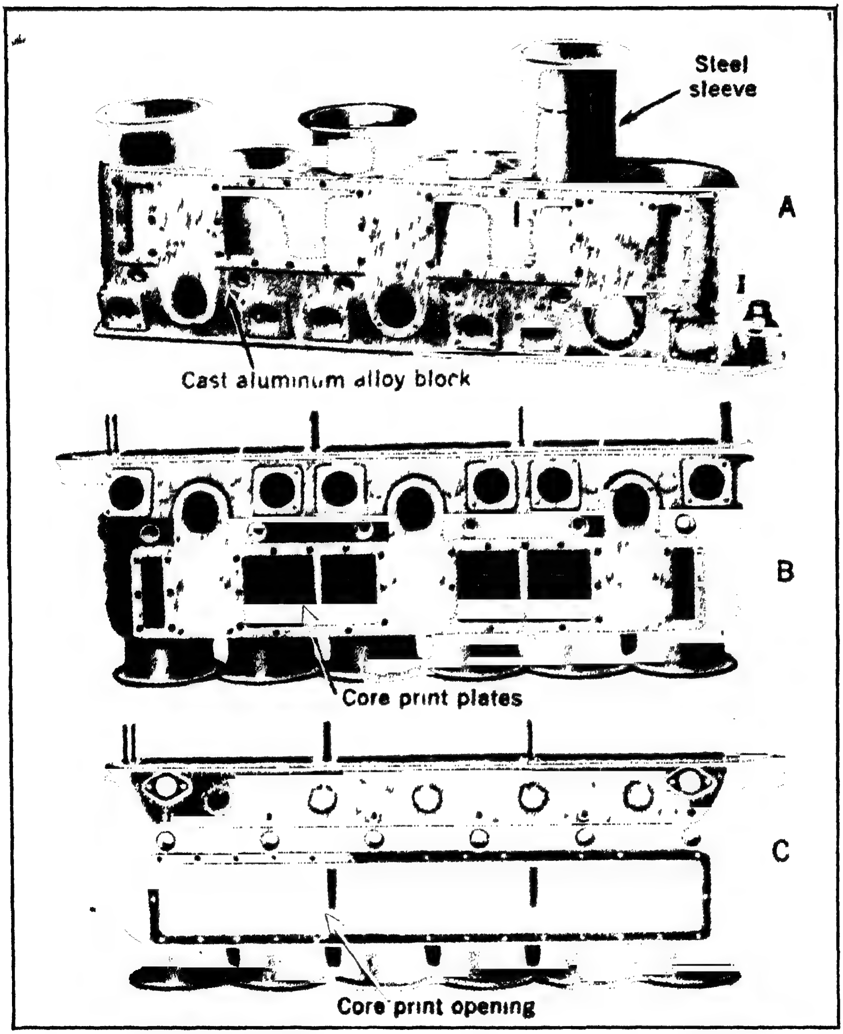


Fig. 247.—Illustration Showing Method of Casting Six-Cylinders and Integral Water Jacket in Aluminum Alloy Used by Hispano-Suiza Engine Manufacturers. Note Inserted Steel Sleeve Forming Retention Members, Because the Base Flange is Part of the Liner. View at A Shows Two Sleeves Pressed in Place, and One Partly in Position. The Side Views at B and C Show the Large Openings Available for Core Print Support when Casting.

ders for the passage of water. Under such circumstances the cooling effect is not even, and the stresses which obtain because of unequal expansion may distort the cylinder to some extent. The system of construction followed in Hispano-Suiza engines and shown at Fig. 247 is exceptionally good. In this the cylinder and head block is cast of aluminum alloy with liberal openings left in the block to permit of excellent core support when casting and insuring uniform cored out water spaces. The cylinder liners are of steel, screwed in place. The core print support openings are closed

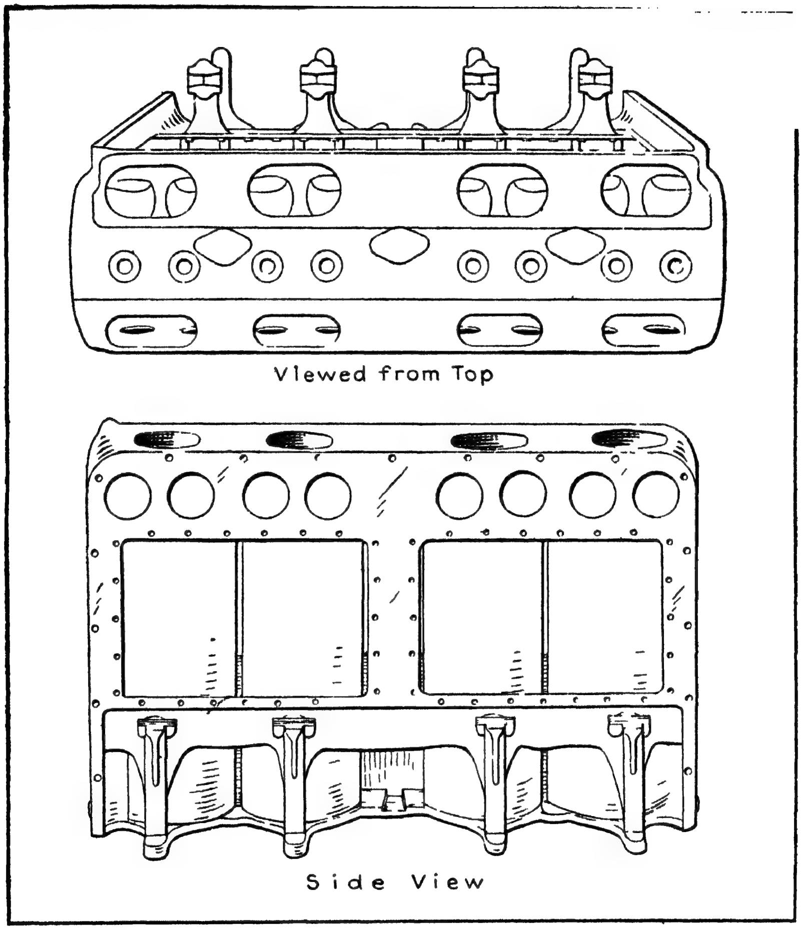


Fig. 248.—Views of Early Four-Cylinder Deusenberg Airplane Engine Cylinder Block.

in by plates when the cylinder is completed. When steel cylinders are made from forgings, the water jackets are usually of copper or sheet steel attached to the forging by autogenous welding; in the case of the latter and, in some cases, the former may be electro-deposited on the cylinders or brazed in place.

Block Castings.—The advantage of casting the cylinders in blocks is that a motor may be considerably shorter than it would be if individual

castings were used. It is admitted that when the cylinders are cast together a more compact, rigid, and stronger powerplant is obtained than when cast separately. There is a disadvantage, however, when the cylinder liners are cast integral with the block or of the same material, as when cast iron is used, because if one cylinder becomes damaged it will be necessary to replace the entire unit, which means scrapping three or more good cylinders because one of the assembly has failed. When the cylinders are cast separately one need only replace the one that has become damaged. The casting of four or more cylinders in one unit is made possible by improved foundry methods, and when proper provision is made for holding the cores when the metal is poured and the cylinder casts are good, the construction is one of distinct merit. It is sometimes the case that the proportion of sound castings is less when cylinders are cast in block, but if the proper precautions are observed in moulding and the proper mixtures of cast iron or aluminum alloy are used, the ratio of defective castings is said to be but slightly more than when cylinders are moulded individually. As an example of the courage of engineers in departing from old-established rules, the cylinder casting shown at Fig. 248 may be considered typical. This was used on the Duesenberg four-cylinder sixteen-valve 4¾-inch by 7-inch engine which had a piston displacement of 496 cubic inches. At a speed of 2,000 r.p.m., corresponding to a piston speed of 2,325 feet per minute, the engine was guaranteed to develop 125 horsepower. The weight of the model engine without gear reduction was 436 pounds, but a number of refinements were made in the design whereby it was expected to get the weight down to 390 pounds. The four cylinders were cast from semi-steel in a single block, with integral heads. The cylinder construction is the same as that which has always been used by Mr. Duesenberg, inlet and exhaust valves being arranged horizontally opposite each other in the head. There are large openings in the water jacket at both sides and at the ends, which are closed by means of aluminum covers, water-tightness being secured by the use of gaskets. This results in a saving in weight because the aluminum covers can be made considerably lighter than it would be possible to cast the jacket walls, and, besides, it permits of obtaining a more nearly uniform thickness of cylinder-wall, as the cores can be much better supported. The cooling water passes completely around each cylinder, and there is a very considerable space between the two central cylinders, this being made necessary in order to get the large bearing area desirable for the central bearing.

It is common automotive engine practice to cast the water jackets integral with the cylinders, if cast iron or aluminum is used, and this is also the most economical method of applying it because it gives good results in practice. In aviation engines, however, the use of cast iron or semi-steel cylinder blocks is now obsolete and where block castings are used, they are of aluminum alloy with inserted cylinder liners.

An important detail is that the water spaces must be proportioned so that they are equal around the cylinders whether these members are cast individually, in pairs, threes, fours or sixes. When cylinders are cast in block form it is good sound practice to leave one or more large openings in the jacket wall which will assist in supporting the core and make for uni-

form water space. It will be noticed that the casting shown at Fig. 248 has a large opening in the side of the cylinder block. These openings are closed after the interior of the casting is thoroughly cleaned of all sand, core wire, etc., by brass, stamped steel or aluminum plates, the latter being preferred. These also have particular value in that they may be removed after the motor has been in use, thus permitting one to clean out the interior of the water jacket and dispose of the rust, sediment, and incrustation which are always present after the engine has been in active service for a time and which interfere with proper thermal conductivity of the heat from cylinder metal to the cooling liquid.

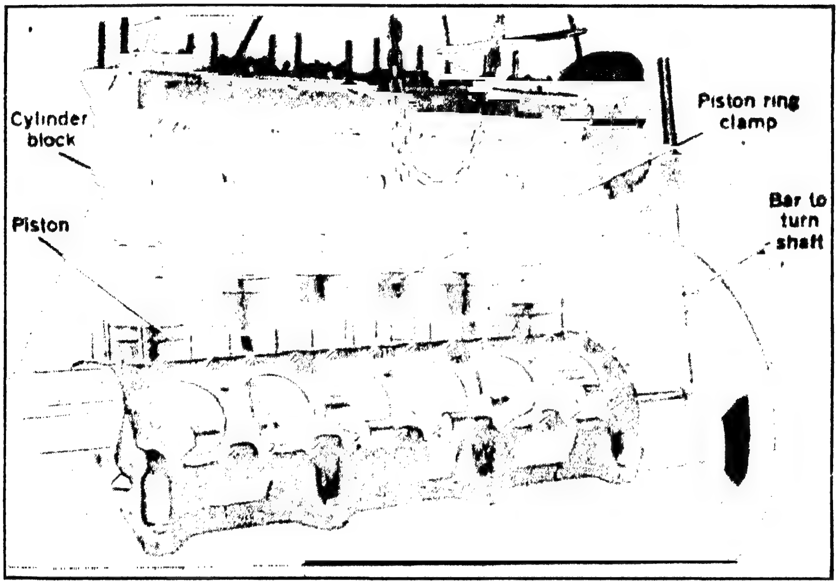


Fig. 249.—Showing Method of Installing a Six-Cylinder Block Casting on Curtiss D12 Engine.

Among the advantages claimed for the practice of casting cylinders in blocks may be mentioned compactness, lightness, rigidity, simplicity of water piping, as well as permitting the use of simple forms of inlet and exhaust manifolds because the gas passages may be cored into the casting. The light weight is not only due to the reduction of the cylinder mass but because the block construction permits one to lighten the entire motor. The fact that all cylinders are cast together decreases vibration, and as the construction is very rigid, disalignment of working parts is practically eliminated. When inlet and exhaust manifolds are cored in the block casting, as is sometimes the case, but one joint is needed on each of these instead of the multiplicity of joints which obtain when the cylinders are individual castings. The water piping is also simplified. In the case of a four- or six-cylinder block motor but two pipes are used; one for the water to enter the cylinder jacket, the other for the cooling liquid to discharge through. In aviation engine practice, most designers favor the individual cylinder of built up or composite construction though there are several notable exam-

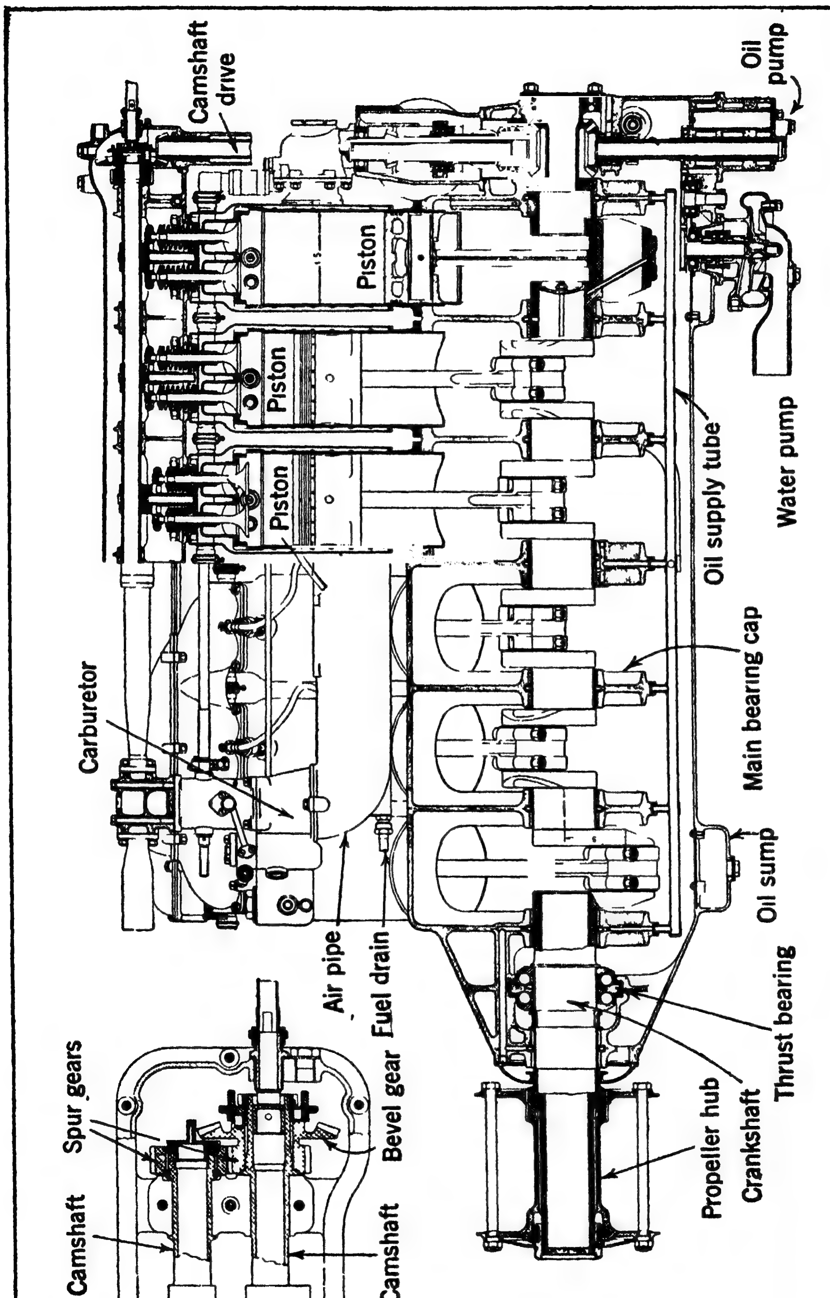


Fig. 250.—Longitudinal Section of Fiat A20 Twelve-Cylinder "Vee" Engine, Showing Method of Cylinder Construction and Valve Actuation.

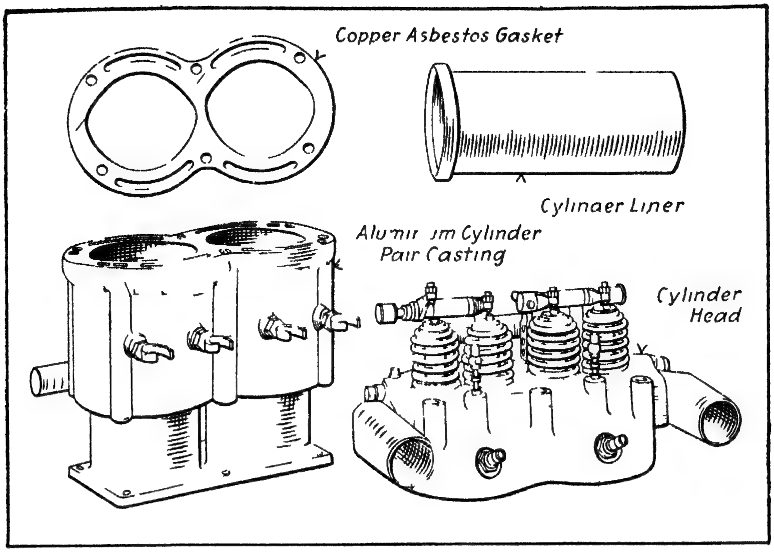


Fig. 251.—Twin-Cylinder Block of Early Sturtevant Engine was Cast of Aluminum and Had Removable Cylinder Heads and Inserted Cylinder Liners.

ples of the use of cast aluminum water jacket and cylinder head blocks and inserted steel barrels, which after assembly are handled just as an auto engine cylinder block casting would be as shown at Fig 249

Influence on Crankshaft Design.—The method of casting or grouping the cylinders has a material influence on the design of the crankshaft as will be shown in proper sequence When four cylinders are combined in one block in automobile engines it is possible to use a two-bearing crankshaft or a three-bearing shaft Where cylinders are cast in pairs a three-bearing crankshaft is commonly supplied, and when four cylinders cast as

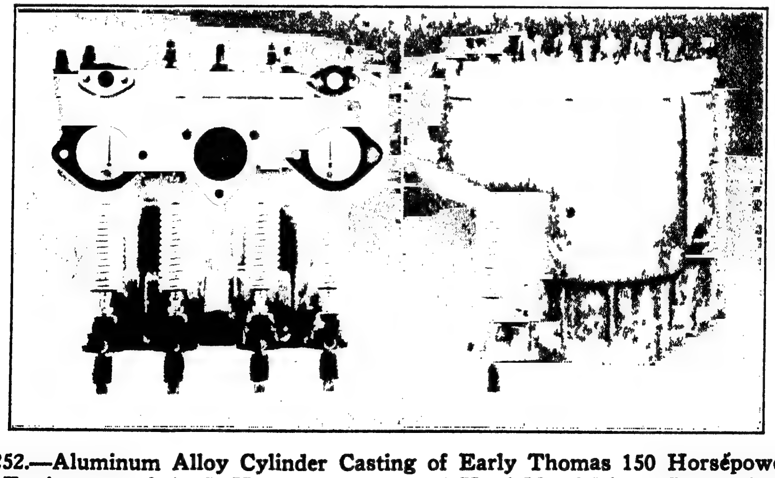


Fig. 252.—Aluminum Alloy Cylinder Casting of Early Thomas 150 Horsepower Airplane Engine was of the L Head Type and Had Hard Metal Liners Inserted to Take Piston Wear.

individual units are used it is thought necessary to supply a five-bearing crankshaft, though sometimes shafts having but three journals are used successfully. Obviously the shafts must be stronger and stiffer to withstand the stresses imposed if two supporting bearings are used than if a larger number are employed. In this connection it may be stated that there is less difficulty in securing alignment with a lesser number of bearings and there is also less friction. On the other hand, the greater the number of points of support a crankshaft has the lighter the webs can be made and still have requisite strength. It is common practice in aviation engines to

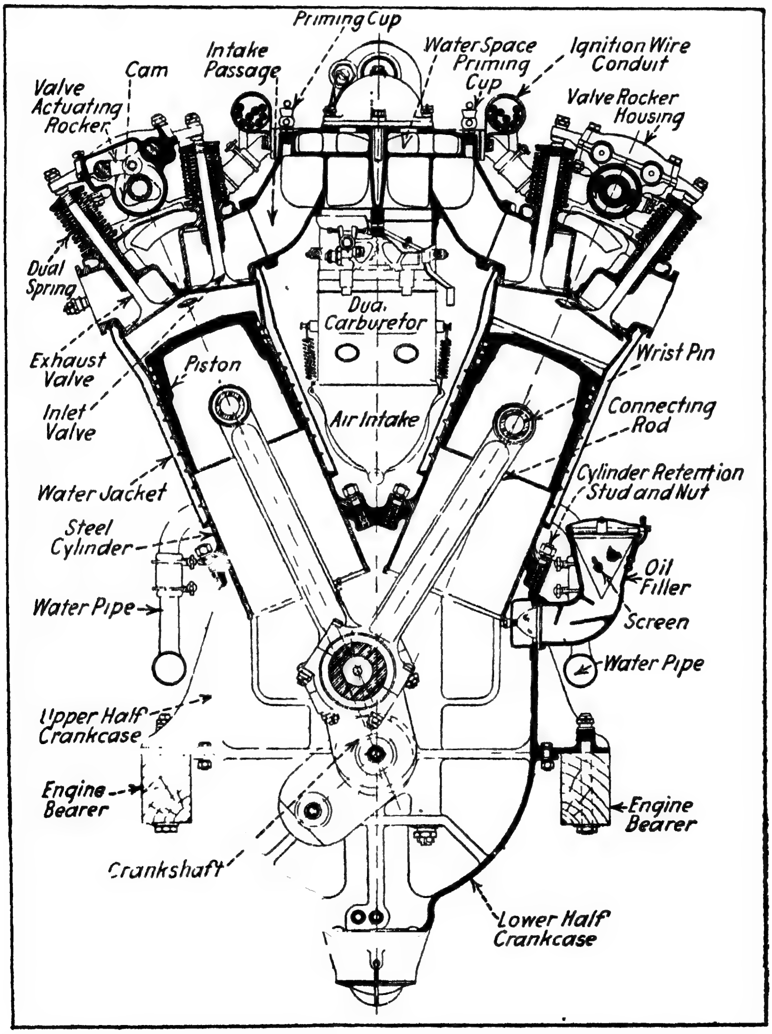


Fig. 253.—Transverse Sectional View Through Cylinders of Liberty Airplane Engine, a Good Example of the "Vee" Water-Cooled Type Having Overhead Valves. This Engine is an Obsolescent Design.

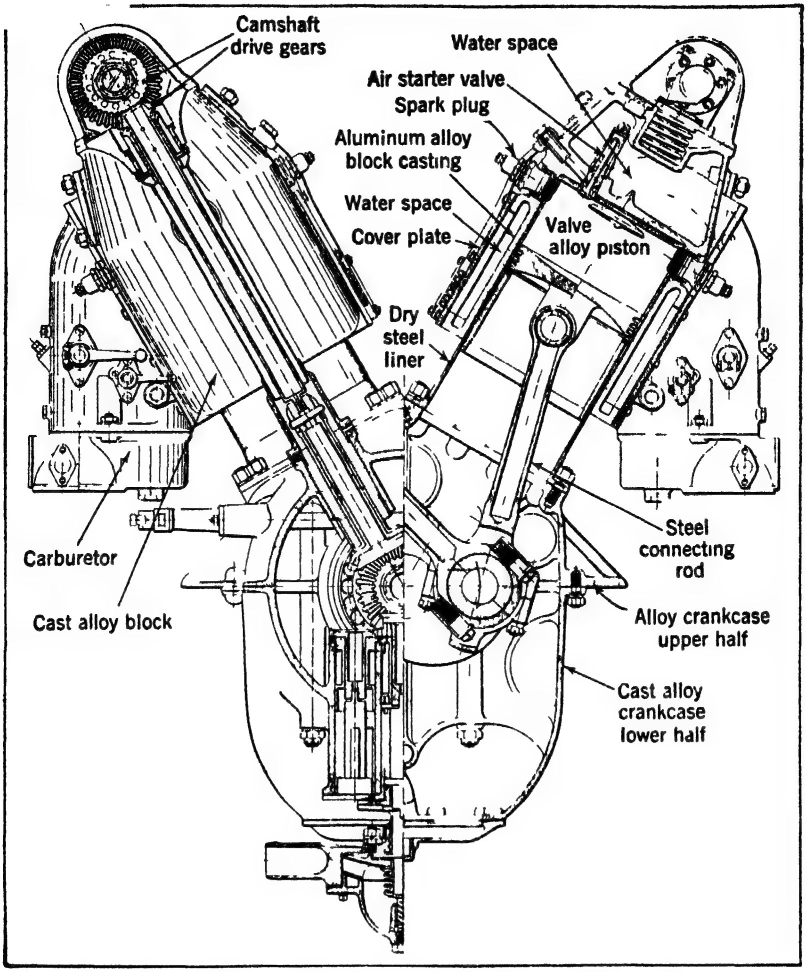


Fig. 254.—Transverse Sectional View of Hispano-Suiza Aviation Engine, Showing How Steel Liner is Installed in Cast Aluminum Alloy Cylinder Block. This Type of Construction is Known as a "Dry Liner" Because no Water Comes in Contact with it. Note, also, Crankcase Construction.

use the individual cylinder construction and this calls for using a bearing between each pair of cylinders, as shown at Fig. 250, which is a longitudinal sectional view of the Fiat A20 engine, in order to use a light crankshaft which is essential in such motors.

Combustion-Chamber Design.—Another point of importance in the design of the water-cooled cylinder, and one which has considerable influence upon the power developed, though not to as great a degree as in air-cooled cylinders, is the shape of the combustion-chamber. The endeavor of designers is to obtain maximum power from a cylinder of certain proportions, and the greater energy obtained without increasing piston displacement or fuel consumption the higher the efficiency of the motor. To prevent troubles

due to preignition it is necessary that the combustion-chamber be made so that there will be no roughness, sharp corners, or edges of metal which may remain incandescent when heated or which will serve to collect carbon deposits by providing a point of anchorage. With the object of providing an absolutely clean combustion-chamber some designers used a separable head unit to their twin-cylinder castings, such as shown at Fig. 251 and Fig. 252. These permit one to machine the entire interior of the cylinder and combustion-chamber. The relation of valve location and combustion-chamber design will be considered in proper sequence. These cylinders were cast of aluminum, instead of cast iron, as was customary in contemporary auto-

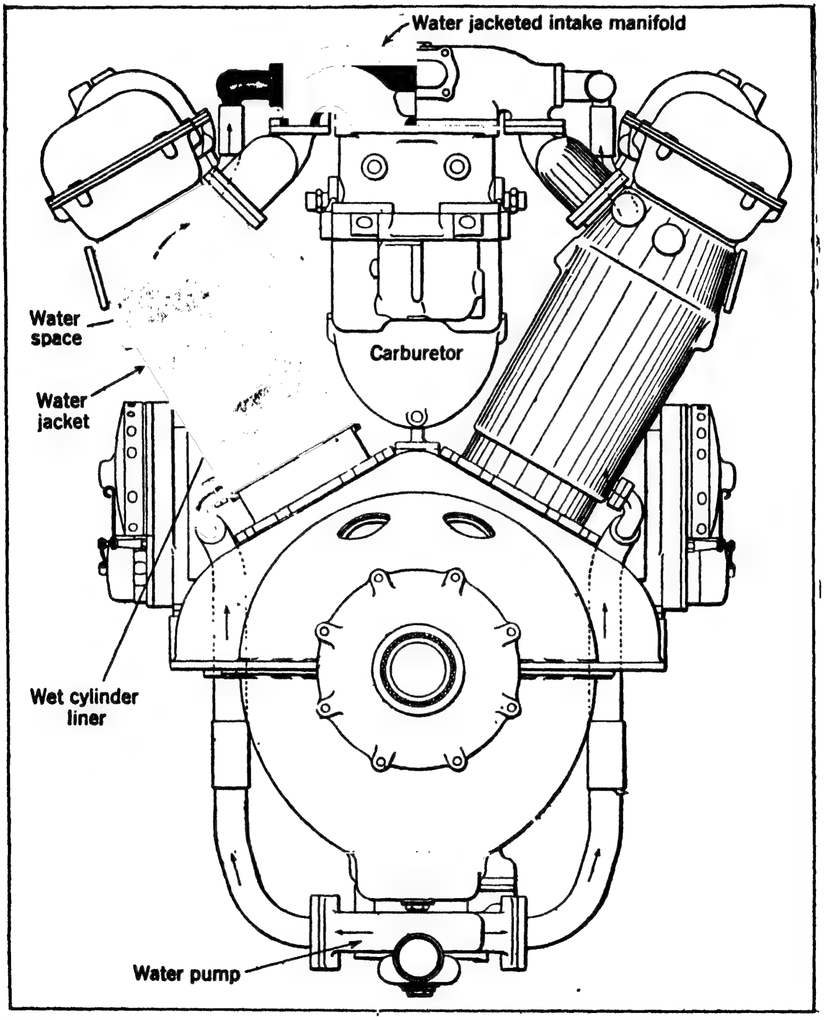


Fig. 255.—Transverse Part-Sectional View of Fiat A20 Aviation Engine Showing the Wet Liner Type of Cylinder Construction. Note Uniform Water Space Possible by Using Applied Water Jackets.

mobile engines and were provided with steel or cast iron cylinder liners forced in the soft metal casting bores. The construction outlined is now obsolete and practically all aviation engines of recent development use steel forged cylinders with applied water jackets to secure maximum lightness as shown in the sectional view of the Liberty engine at Fig. 253.

Water-Cooled Cylinder Development.—There are two generic types from which modern water-cooled cylinders have been evolved, the dry sleeve and the wet sleeve types. The early cylinder construction used on Hispano-Suiza engines, consisted of flanged, closed end steel sleeves threaded into an aluminum block, as shown at Fig. 254. It has been stated that this engine could not be operated for more than 30 hours at full throttle without valve grinding. An improved cylinder construction was used on the Wright E3 motor and is similar to the form used on the E2 engine except that no threads are used on the sleeve, which is a force shrink fit in the aluminum water jacket block. The sleeves were held in position by studs in the top of the sleeve which passed through the top of the block.

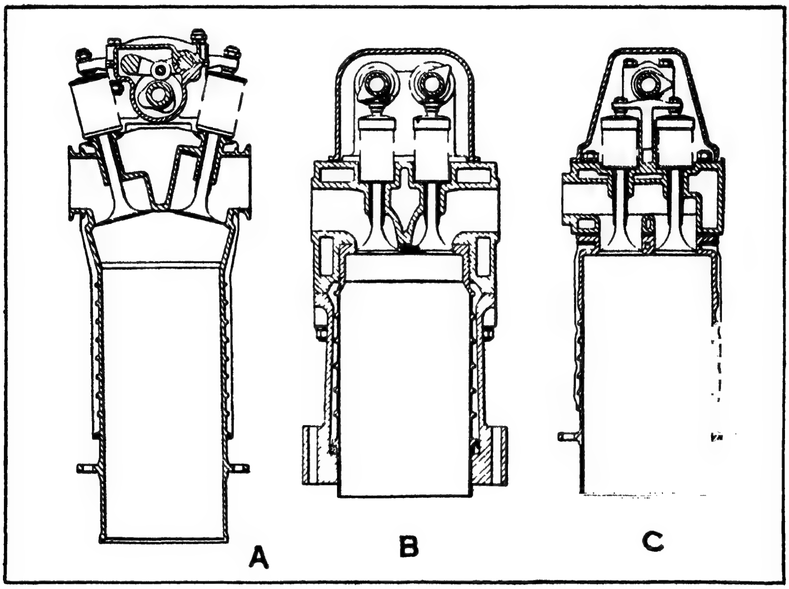


Fig. 256.—Forms of Cylinders Used in Practical Water-Cooled Aircraft Engines. A—Cylinder of Liberty Engine. B—Curtiss Wet Sleeve Construction. C—Packard Cylinder Construction.

This construction was not as good theoretically as the threaded sleeve because not as much surface was in contact with the aluminum jacket wall, but it gave good results in practice. In all these types, the valves seated in the top of the steel sleeve as shown. The last form to be used in Wright engines was one in which the cylinder sleeve is screwed tightly into a cast aluminum alloy cylinder block which has the heads cast integrally. The valves seat into bronze inserts in the aluminum alloy head just as in air-cooled composite cylinder practice. The sleeve screws tightly into the block and seats against a shoulder in the aluminum combustion head. This

method of construction is known as the "dry sleeve" cylinder, and was originally used by the designer of Hispano-Suiza engines, Mr. Marc Birkigt, a Swiss engineer associated with a Spanish automobile manufacturer. It is called the "dry sleeve" because the water did not come into direct contact with the cylinder liner as in the form known as the "wet sleeve" and shown at Fig. 255. The cooling effect is more direct in a "wet sleeve" cylinder than in a "dry sleeve" type.

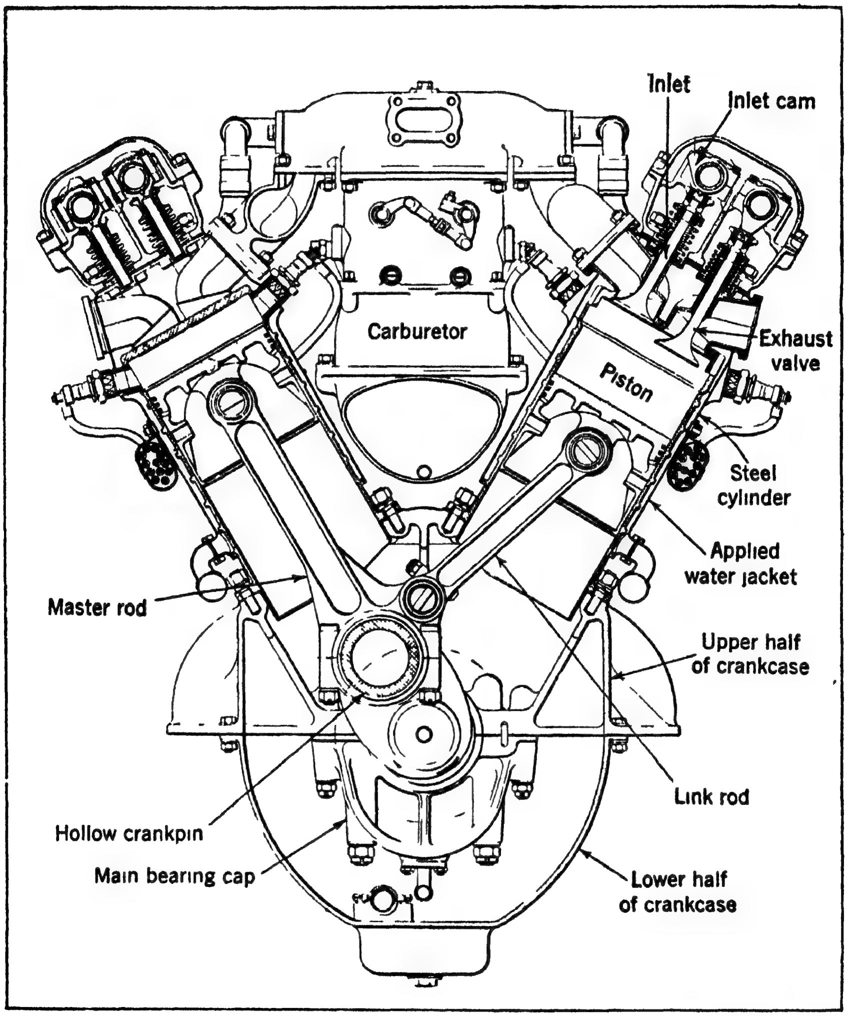


Fig. 257.—Transverse Sectional View of Fiat A20 Aviation Engine, Showing Method of Valve Actuation by Side by Side Camshafts and also Outlining Clearly Cylinder and Crankcase Construction.

Wet Sleeve Construction.—The other method of cylinder construction is called the "wet sleeve" and various cylinders have been experimented with as shown at Fig. 256. That at A is the well-known Liberty engine cylinder. The design shown at B is a Curtiss construction in which a closed

end sleeve is threaded into an aluminum combustion head block, and the water jacket is completed by another casting bolted to the combustion head. A watertight joint is obtained by a ring of packing material at the bottom of the sleeve. Attention is directed to the heavy section needed around the bolts at the retention flange when aluminum alloy is used instead of steel for the water jacket block. The cylinder shown at Fig. 256 C is the final type evolved as a result of much study and practical testing by the Packard engineers. Each block is composed of six individual cylinders attached to a single aluminum casting that is termed the valve-housing. The individual cylinder is composed of a drawn-steel sleeve welded to a forged combustion-chamber head machined completely, and having a head-plate and a sheet-metal water jacket welded into place. Each cylinder is provided with four valves, short valve-ports being formed integral with the cylinder. These valve-ports are accurately hollow-milled on their outer surfaces; and the head-plate is bored so as to form a pressfit over the valve-ports, the plate seating on shoulders so as to provide about three-eighths inch water-space above the combustion-chamber. The cylinder head is provided with five bosses into which are screwed long studs for supporting the valve-housing. The sparkplug bosses are formed integral with the combustion-chamber. The cylinder retention flange is so placed that the cylinders project into the crankcase. A European modification of the "wet sleeve" construction of the Liberty cylinder is shown at Fig. 257, which is a transverse section of the Fiat A20 aviation engine. This cylinder differs from the Liberty in the form of the combustion-chamber and in the system of valve actuation, the latter being the method used in this country on Curtiss D12 engines.

Valve Location.—It has often been said that a chain is no stronger than its weakest link, and this is as true of the explosive motor as it is of any other piece of mechanism. Many motors which appear to be excellently designed and which were well constructed did not prove satisfactory because some minor detail or part had not been properly considered by the designer. A factor having material bearing upon the efficiency of either the air- or water-cooled internal-combustion motor is the location of the valves and the shape of the combustion-chamber which is largely influenced by their placing. The fundamental consideration of high-speed engine valve design is that the gases be admitted and discharged from the cylinder as quickly as possible in order that the speed of gas flow will not be impeded and produce back-pressure. This is imperative in obtaining satisfactory operation in any form of motor. If the inlet passages are constricted the cylinder will not fill with explosive mixture promptly, whereas if the exhaust gases are not fully expelled the parts of the inert products of combustion retained dilute the fresh charge, making it slow burning and causing lost power and overheating. When an engine employs water as a cooling medium this substance will absorb the surplus heat readily, and the effects of overheating are not noticed as quickly as when air-cooled cylinders are employed. Valve sizes have a decided bearing upon the speed of motors and some valve locations permit the use of larger members than do other positions.

While piston velocity is an important factor in determinations of power output, it must be considered from the aspect of the wear produced upon

the various parts of the motor. It is evident that engines which run very fast, especially of high power, must be under a greater strain than those operating at lower speeds. The valve-operating mechanism is especially susceptible to the influence of rapid movement, and the slower the engine the longer the parts will wear and the more reliable the valve action.

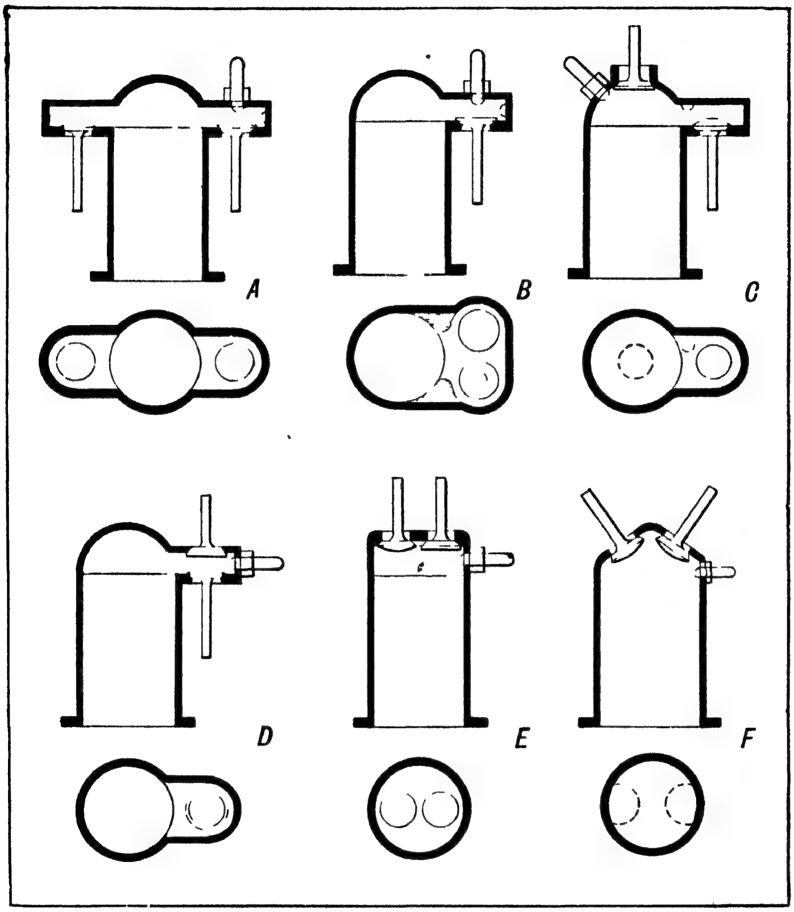


Fig. 258.—Diagrams Showing Forms of Cylinder and Combustion-Chambers Demanded by the Valve Placing. A—Tee Head Type, Valves on Opposite Sides. B—L Head Cylinder, Valves Side by Side. C—L Head Cylinder; One Valve in Head, the Other in Side Pocket. D—Inlet Valve Over Exhaust Member, Both in Side Pocket. E—Valve in Head Type with Vertical Valves. F—Inclined Valves Placed to Open Directly into Combustion-Chamber.

T Head Cylinder.—As will be seen by reference to the accompanying illustration, Fig. 258, there are many ways in which valves may be placed in the cylinder. Each method outlined possesses some point of advantage, because all of the types illustrated are used by reputable automotive engine manufacturers The method outlined at Fig. 258 A is now seldom used, and because of its shape the cylinder is known as the "T" form. It was approved for automobile and marine engine use for several reasons, the

most important being that large valves can be employed and a well-balanced and symmetrical cylinder casting obtained. Two independent camshafts are needed, one operating the inlet valves, the other the exhaust members. The valve-operating mechanism can be very simple in form, consisting of a plunger actuated by the cam which transmits the cam motion to the valve-stem, raising the valve as the cam follower rides on the point of the cam. Piping may be placed without crowding because only one pipe assembly need be placed on each side and larger manifolds can be fitted than in some other constructions. This has special value in large marine installations as it permits the use of an adequate discharge pipe on the exhaust side with its obvious advantages. This method of cylinder construction is never found on airplane engines because it does not permit of maximum power output though it does have a high volumetric efficiency.

If considered from a viewpoint of actual heat efficiency, it is theoretically the worst form of combustion-chamber. This disadvantage is probably compensated for to some extent by uniformity of expansion of the cylinder because of balanced design. The ignition sparkplug may be located directly over the inlet valve in the path of the incoming fresh gases, and both valves may be easily removed and inspected by unscrewing the valve caps without taking off the manifolds. Carbon removal is also possible without calling for a separable head casting as scrapers are easily introduced through the large ports. This is an advantage of some importance in large marine engines.

The valve installation shown at C is somewhat unusual, though it provides for the use of valves of large diameter. Easy charging is insured because of the large inlet valve directly in the top of the cylinder. Conditions may be reversed if necessary, and the gases discharged through this large valve. Both methods have been used, though it would seem that the free exhaust provided by allowing the gases to escape directly from the combustion-chamber through the overhead valve to the exhaust manifold would make for more power. In the Hudson automobile engine of most recent design, this construction is used though the inlet valve is placed in the head while the exhaust valve is in the side pocket. The method outlined at Fig. 258 F and at Fig. 253 is one that has been widely employed on large automobile racing motors where extreme power is required, as well as in engines constructed for aviation service. The inclination of the valves permits the use of large valves, and these open directly into the combustion-chamber. There are no pockets to retain heat or dead gas, and free intake and outlet of gas is obtained. This form is quite satisfactory from a theoretical point of view because of the almost ideal combustion-chamber form. Some difficulty is experienced, however, in properly water-jacketing the valve chamber which experience has shown to be necessary if the engine is to have high power and to be of the water-cooled form.

L Head Cylinder.—The motor shown at Fig. 258 B and Fig. 252 employs cylinders of the "L" type. Both valves are placed in a common extension from the combustion-chamber, and being located side by side both are actuated from a common camshaft. The inlet and exhaust pipes may be placed on the same side of the engine and a very compact assemblage is obtained, though this is optional as the manifolds may be placed on opposite sides if

Column No. Type	1 Type of Cylinder Head	2 Form of Head and Position of Spark Plugs	3 Valves per Cylinder	4 Cylinders	5 Volume of Cylinder, cu in	6 Compression-Ratio	7 Speed, r p m.	8 Brake Mean Pressure, lb per sq in	9 Mechanical Efficiency, per cent	10 Indicated Mean Effective Pressure, lb per sq in	11 Indicated Mean Effective Pressure Corrected for R=5, lb per sq in	12 Indicated Mean Effective Pressure Corrected for 100 cu in of Cylinder Volume, lb per sq in	13 Indicated Thermal Efficiency Corrected for R=5, per cent	14 Relative Efficiency, per cent	Remarks
Stationary *	Plain Cylindrical		2	1	28.95	4.45	1200	110	84.5	130.0	135.5	137.5	30.0	63.3	Slight Tendency to Detonate
Experimental Stationary *	Plain Cylindrical		5	1	126.90	5.00	1750	121	86.5	140.0	140.0	139.5	32.0	67.5	
Automobile *	Plain Cylindrical		2	2	30.80	4.80	1750	114	83.0	137.5	140.0	142.0	30.4	64.0	
Aeronautic *	Plain Cylindrical		2	6	106.70	4.70	1400	121	87.0	139.0	143.0	143.0	27.3	57.5	
Racing Motorcycle *	Pent Roof		4	1	30.40	5.40	3000	121	87.5	139.0	135.0	137.0			No Detonation under any Circumstances
Experimental *	Pent Roof		4	1	[illegible]	4.85	1250	135	88.0	153.0	155.0	143.0	33.0	69.5	Slight Tendency to Detonate
Aeronautic *	Pent Roof		4	12	72.60	4.80	2200	120	88.0	136.5	138.5	139.0	27.7	58.3	
Experimental *	Ⴈ Head		2	1	83.00	5.40	1500	114	87.0	131.0	127.0	127.0	31.4	66.2	Excessive Detonation, could only be run on Benzol
Tank *	Ⴈ-Head		2	6	175.50	4.30	1200	106	87.0	122.0	129.0	127.0	31.1	65.5	Severe Detonation
Motor Truck *	Ⴈ-Head		2	4	101.20	4.70	1200	102	83.0	123.0	126.0	127.0	29.2	61.5	Severe Detonation
Automobile *	Special Side Valve		2	4	35.40	5.00	1800	107	88.0	122.0	122.0	124.0	28.1	59.2	Very Little Detonation
Motor Truck *	Special Side Valve		2	4	99.20	4.70	1100	104	88.0	118.0	121.5	121.5	28.3	59.6	
Automobile *	Special Side Valve		2	4	23.80	4.80	2000	96	86.0	112.0	114.5	117.5	29.9	63.0	
Automobile *	Special Side Valve		2	4	31.70	5.00	1800	109	86.0	127.0	127.0	128.5	31.1	65.5	
Automobile	Ordinary Type Side Valve		2	4	35.40	5.05	1600	96	87.0	110.0	110.0	112.0	24.2	51.0	Severe Detonation
Automobile	Ordinary Type Side Valve		2	4	99.20	3.40	1000	86	86.0	100.0	114.0	114.0	26.9	56.6	
Automobile	Ordinary Type Side Valve		2	4	23.80	4.80	2000	90	86.0	105.0	110.0	113.0	26.9	56.6	
Automobile	Ordinary Type Side Valve		2	4	50.30	4.00	1750	92	87.0	106.0	115.0	116.0	24.9	52.5	
Motor Truck *	T-Head		2	4	94.60	4.25	1400	94	90.5	104.0	112.0	112.0	26.4	56.6	No Detonation
Motor Truck	T-Head		2	4	95.20	4.00	1200	76	81.0	94.0	102.0	102.0	25.3	53.3	Severe Detonation

Fig. 259A.—Results of a Series of Tests Tabulated by H. R. Ricardo Showing Influence of Combustion-Chamber Form on Efficiency, Detonation, etc.

passages are cored in the cylinder pairs to lead the gases to opposite sides. The valves may be easily removed if desired, and the construction is fairly good from the viewpoint of both foundry man and machinist. The chief disadvantage is the limitation to the area of the valves and the loss of heat efficiency due to the pocket. This form of combustion-chamber, however, is more efficient thermally than the "T" head construction, though with the latter the use of larger valves probably compensates for the greater heat loss. It has been stated as an advantage of this construction that both

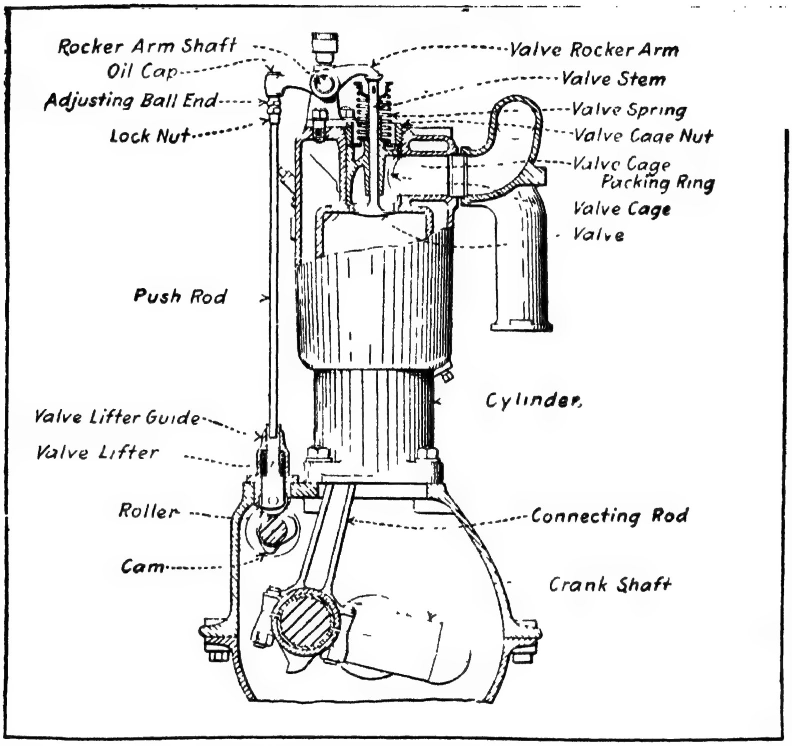


Fig. 259.—Sectional View of Early Buick Automotive Engine Cylinder, Showing Valve and Cage Installation. A Construction Now Used Only on Large Marine and Stationary Engines.

manifolds can be placed at the same side of the engine and a compact assembly secured. On the other hand, the disadvantage may be cited that in order to put both pipes on the same side they must be of smaller size than can be used conveniently when the valves are oppositely placed. The "L" form cylinder is sometimes made more efficient if but one valve is placed in the pocket while the other is placed over it. This construction is well shown at Fig. 258 D and was found on early Anzani motors.

The method of valve application shown at Fig. 253 is an ingenious method of overcoming some of the disadvantages inherent with valve-in-the-head motors. In the first place it is possible to water-jacket the valves thoroughly, which is difficult to accomplish when they are mounted in cages as is sometimes done in other automotive applications distinct from

aircraft as illustrated at Fig. 259. The water circulates directly around the walls of the valve chambers, which is superior to a construction where separate cages are used, as there are two thicknesses of metal with the latter, that of the valve-cage proper and the wall of the cylinder. The cooling medium is in contact only with the outer wall, and as there is always a loss of heat conductivity at a joint it is practically impossible to keep the exhaust valves and their seats at a uniform temperature. The valves may be

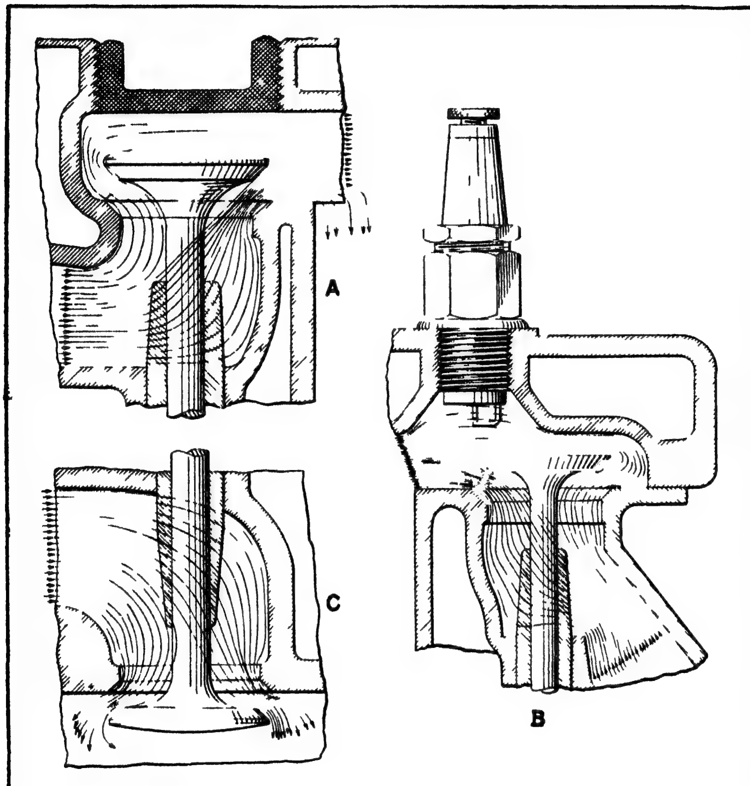


Fig. 260.—Diagrams Showing How Gas Enters Cylinder Through Varying Valve Types. A—"Tee" Head Cylinder. B—L Head Cylinder. C—Overhead Valve.

of larger size without the use of pockets when seating directly in the head as in the construction shown at Fig. 251. In fact, they could be equal in diameter to almost half the bore of the cylinder, which provides an ideal condition of charge placement and exhaust. When valve grinding is necessary the entire head is easily removed by taking off some nuts and loosening inlet manifold connections, and a similar operation would be necessary even if cages were employed, as in the engine shown at Fig. 259, because the cages must be removed.

I Head Cylinders.—At Fig. 260 A and B a section through a typical "L"-shaped cylinder is depicted It will be evident that where a pocket construction is employed, in addition to its faculty for absorbing heat, the passage

of gas would be impeded. For example, the inlet gas rushing in through the open valve would impinge sharply upon the valve-cap or combustion head directly over the valve and then must turn at a sharp angle to enter the combustion-chamber and then at another sharp angle to fill the cylinders. The same conditions apply to the exhaust gases, though they are reversed. When the valve-in-the-head type of cylinder is employed, as at C, the only resistance offered the gas is in the manifold. As far as the

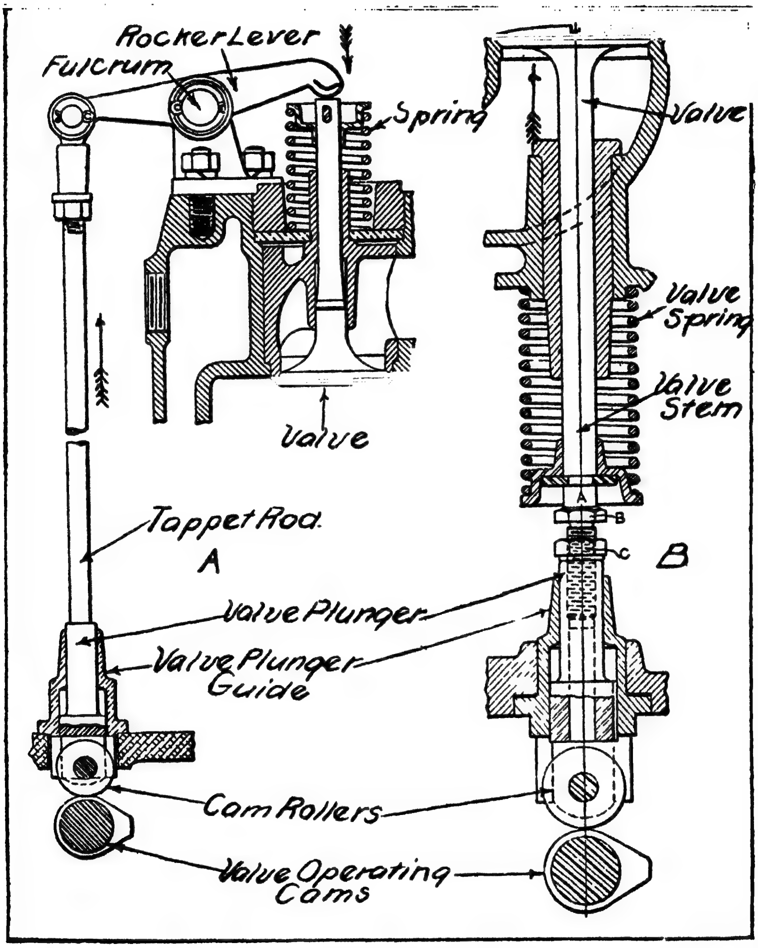


Fig. 261.—Conventional Methods of Operating Internal-Combustion Motor Valves. A—Overhead Valve. B—Valve in Side Pocket.

passage of the gases in and out of the cylinder is concerned, ideal conditions obtain. It is claimed that valve-in-the-head motors are more flexible and responsive than other forms, but the construction has the disadvantage in that the valves must be opened through a rather complicated system of push rods and rocker arms if the camshaft is mounted in the engine base instead of the simpler and direct plunger which can be used with either the

"T" or "L" head cylinders. This is clearly outlined in the illustrations at Fig. 261, where A shows the valve-in-the-head operating mechanism necessary if the camshaft is carried at the cylinder base, while B shows the most direct push-rod action obtained with "T" or "L" head cylinder placing.

The objection can be easily met by carrying the camshaft above the cylinders and driving it by means of gearing. The types of engine cylinders using this construction are shown at Fig. 262 and it will be evident that a positive and direct valve action is possible by following the construction originated by the Mercedes (German) aviation engine designers and outlined at A. The other forms at B and C are very clearly adaptations of this design as are the Liberty motor and Lorraine engine valve actions. The

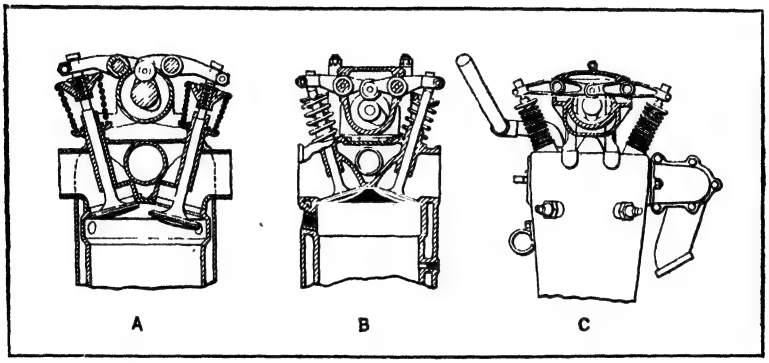


Fig. 262.—Examples of Direct Valve Actuation by Overhead Camshafts. A—The Original Design Used on Mercedes Engines. B—Hall-Scott. C—Early Wisconsin Engine.

Hispano-Suiza engine at Fig. 263 is depicted in part section and no trouble will be experienced in understanding the bevel pinion and gear drive from the crankshaft to the overhead camshaft through a vertical countershaft. A very direct valve action was used in the Duesenberg engines. The valve stems were parallel with the piston top and were actuated by rocker arms, one end of which bore against the valve stem, and the other against the cam on the camshaft. The difference between the Liberty valve action and that of the Hispano-Suiza engines is in the method of imparting movement from the overhead camshaft to the valve stem. In the Liberty cylinder, shown in its relation to the rest of the engine assembly at Fig. 253, rocker arms are used, one end bearing on the cam, and the other on the valve stem. In the Hispano-Suiza engine shown at Fig. 263, the cams bear directly against large area members that also retain the valve spring screwed to the valve stems. Sometimes two side by side overhead camshafts are used as shown at Figs. 250 and 257, these operating the valve stems by direct cam action. When this construction is employed one camshaft is driven by bevel gearing and drives the other by simple spur gearing as shown in inset in Fig. 250.

Concentric Valves.—The form shown at Fig. 264 shows an ingenious application of the valve-in-the-head idea which permits one to obtain large valves. It has been used on some of the early Panhard aviation engines

and on the now obsolete American Aeromarine powerplants. The inlet passage is controlled by the sliding sleeve which is hollow and slotted so as to permit the inlet gases to enter the cylinder through the regular type poppet valve which seats in the exhaust sleeve. When the exhaust valve is operated by the tappet rod and rocker arm the intake valve is also carried down with it. The intake gas passage is closed, however, and the burned

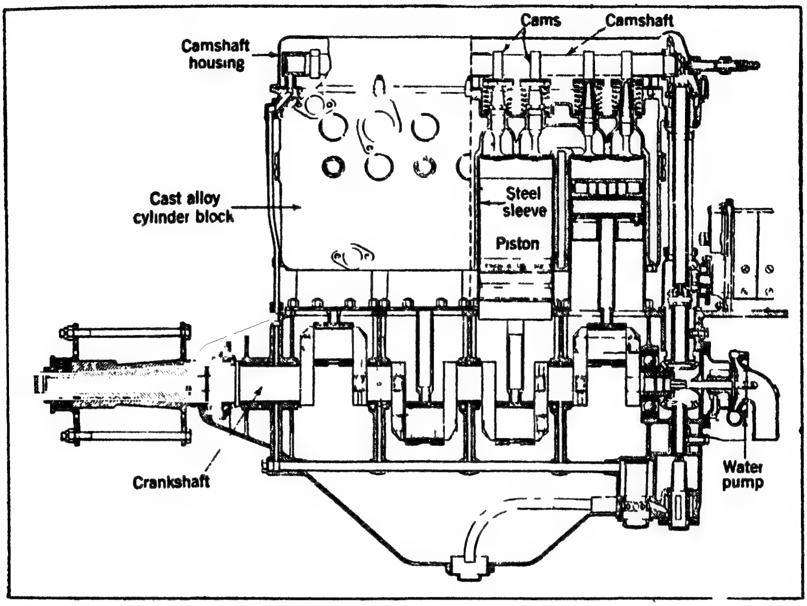


Fig. 263.—Longitudinal Sectional Elevation of Eight-Cylinder "Vee" Engine, Showing Method of Valve Actuation by Cams Acting Directly Against Wear Pieces Carried by Valve Stems. This View also Shows Camshaft Drive and Cylinder Construction.

gases are discharged through the large annular passage surrounding the sleeve. When the inlet valve leaves its seat in the sleeve the passage of cool gas around the sleeve keeps the temperature of both valves to a low point and the danger of warping is minimized. A dome-shaped combustion-chamber may be used, which is an ideal form in conserving heat efficiency, and as large valves may be installed the flow of both fresh and exhaust gases may be obtained with minimum resistance. The intake valve is opened by a small auxiliary rocker arm which is lifted when the cam follower rides into the depression in the cam by the action of the strong spring around the push rod. When the cam follower rides on the high point the exhaust sleeve is depressed from its seat against the cylinder. By using a cam having both positive and negative profiles, a single rod suffices for both valves because of its push and pull action. While a theoretical consideration of the concentric valve application may indicate that there is considerable merit in this design, practical operating experience showed that troubles from warping resulted and that the usual side by side valve construction was the most practical in actual service.

Valve Operation.—The methods of valve operation commonly used vary according to the type of cylinder construction employed In all cases the valves are lifted from their seats by cam-actuated mechanism. Various

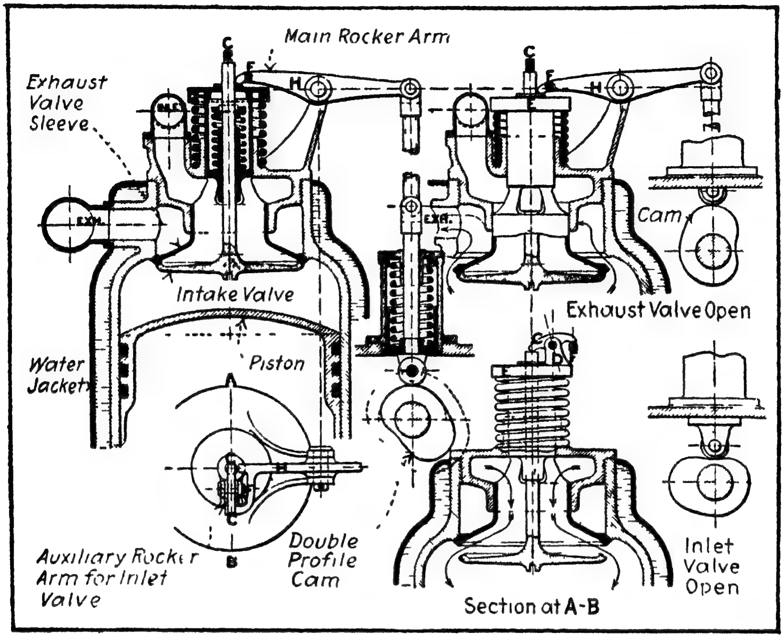


Fig. 264A.—Sectional Views Showing Arrangement and Installation of Novel Concentric Valve Arrangement Devised for Early Panhard Engines.

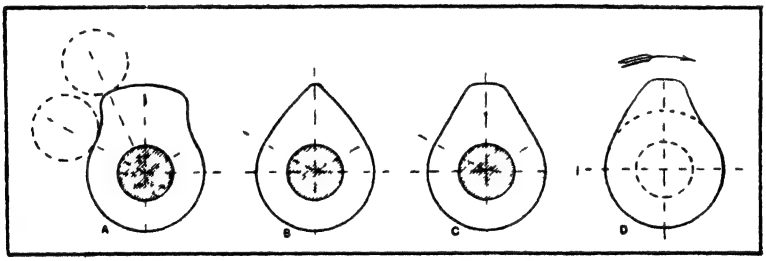


Fig. 264B.—Forms of Valve Lifting Cams Sometimes Employed. A—Cam Provided for Long Dwell and Quick Lift. B—Typical Inlet Cam Used with Mushroom Type Followers. C—Average Form of Cam Used with Roller Followers. D—Profile Designed to Give Quick Lift, Suited Only for Slow-Speed Engines.

forms of valve-lifting cams are shown at Fig. 264. As will be seen, a cam consists of a circle to which a raised, approximately triangular member has been added at one point or as in radial engines where one cam ring may have a number of lobes spaced on its periphery as shown at Fig. 265. When the cam follower rides on the circle, as shown at Fig. 266, there is no difference in height between the cam center and its periphery and there is no movement of the plunger. As soon as the raised portion of the cam strikes

the plunger it will lift it, and this reciprocating movement is transmitted to the valve stem by suitable mechanical connections.

The cam forms outlined at Fig. 264 are some that have been commonly used. That at A is used on engines where it is desired to obtain a quick lift and to keep the valve fully opened as long as possible. It is a noisy form, however, and is not very widely employed except on slow-speed stationary engines. That at B is utilized more often as an inlet cam while the profile shown at C is generally depended on to operate exhaust valves. The cam shown at D is a composite form which has some of the features of the other three types. It will give the quick opening of form A, the gradual closing of form B, and the time of maximum valve opening provided by cam profile C. Cams vary in form according to the valve timing desired and the form of cam follower employed. A cam profile suitable for a roller follower is not at all adapted for one using a mushroom type follower. Most aviation engines use roller followers.

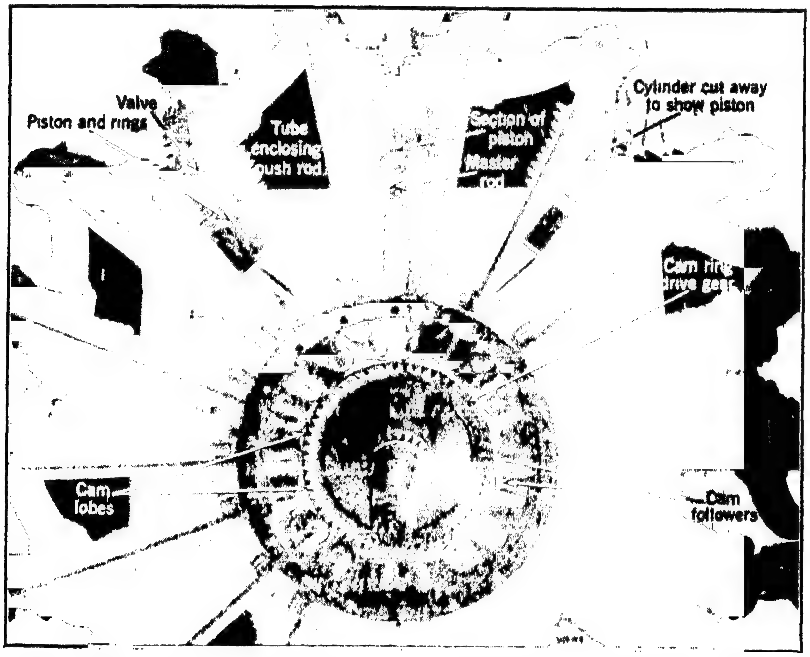


Fig. 265.—View of Timing and Valve Operating Mechanism of the Wright "Whirlwind" Aircraft Engines. Note Roller Cam Followers and Four Lobed Inlet and Exhaust Cam Rings.

The various types of valve followers used are shown at Fig. 266. That shown at A is the simplest form, consisting of a simple square or cylindrical section member having a rounded end which follows the cam profile. These are sometimes made of rectangular stock or if made of round section are kept from rotating by means of a key or pin. A line contact is possible when the plunger is kept from turning, whereas but a single point bearing is obtained when the plunger is cylindrical, has a ball end and is free to revolve. The plunger shown at A will follow only cam profiles which have

gradual lifts. The plunger shown at B is left free to revolve in the guide bushing and is provided with a flat mushroom head which serves as a cam follower. It is widely used in automobile applications but its use is limited in aviation engines. The type shown at C carries a roller at its lower end and may follow very irregular cam profiles if abrupt lifts are desired. While forms A and B are the simplest, that outlined at C in its various forms is more widely used in aviation motors. Compound plungers are used on the Curtiss OX2 motors, one inside the other. The small or inner one works on a cam of conventional design, the outer plunger follows a profile having a flat spot to permit of a pull rod action instead of a push rod action. This engine is shown at Fig. 267.

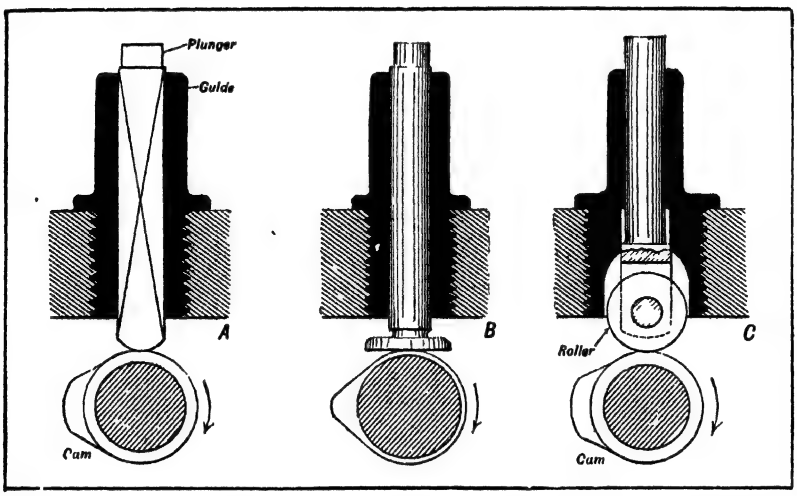


Fig. 266.—Diagram Showing Principal Types of Cam Followers that Have Received General Application in Automotive Engines.

All the methods in which levers are used to operate valves are more or less noisy because clearance must be left between the valve stem and the end of the lever. The space must be taken up before the valve will leave its seat, and when the engine is operated at high speeds the forcible contact between the plunger and valve stem produces a rattling sound until the valves become heated and expand and the stems lengthen out. Clearance must be left between the valve stems and actuating means. This clearance is clearly shown in Fig. 268 and should be .020 inch (twenty thousandths) when engine is cold in the form shown. The amount of clearance allowed depends entirely upon the design of the engine, method of cooling and valve actuation, and length of valve stem. On the Curtiss OX2 engines the clearance is but .010 inch (ten thousandths) because the valve stems are shorter. Too little clearance will result in loss of power or misfiring when engine is hot because the expansion or lengthening of the stem will result in the valve remaining partly open and allowing gas to leak by. Too much clearance will not allow the valve to open its full amount and will disturb the timing and will cause loss of power by reducing volumetric efficiency. The valve clearances needed with various engines will be con-

sidered as each engine is described.

Methods of Driving Camshaft.—Two systems of camshaft operation are used The most common of these is by means of toothed gearing of some form If the camshaft is at right angles to the crankshaft and carried in the engine base it may be driven by worm, spiral, or bevel gearing. If the camshaft is parallel to the crankshaft, simple spur gear or chain connection may

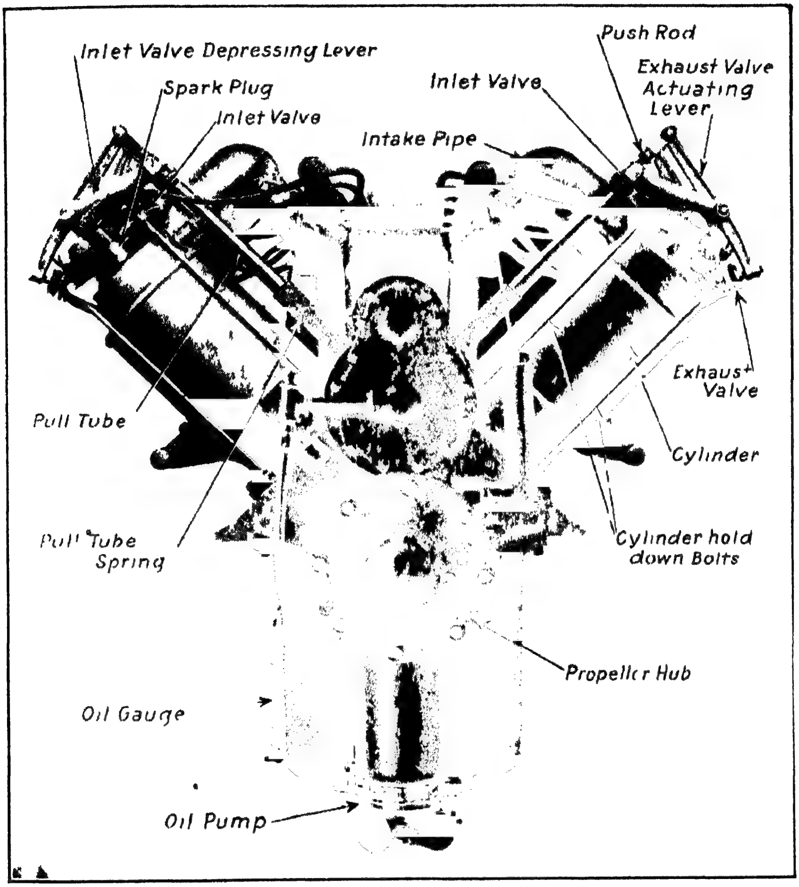


Fig. 267.—Front View of Curtiss OX3 Aviation Motor, Showing Unconventional Valve Action by Concentric Push Rod and Pull Tube.

be used to turn it A typical camshaft mounted in the Vee between cylinders at the top of the crankcase for an eight-cylinder engine is shown at Fig 269. It will be seen that the sixteen cams are forged integrally with the shaft and that it is spur-gear driven. The camshaft drive of the usual aviation motor is shown at Fig 263.

While gearing is more commonly used, considerable attention has been directed of late to silent chains for camshaft operation of automobile motors. The ordinary forms of block or roller chain have not proven successful in this application, but the silent chain, which is in reality a link

belt operating over toothed pulleys, has demonstrated its worth. It first came to public notice when employed on the Daimler-Knight engine for driving the small auxiliary crankshafts which reciprocated the sleeve valves. The advantages cited for the application of chains are, first, silent operation, which obtains even after the chains have worn considerably; second, in

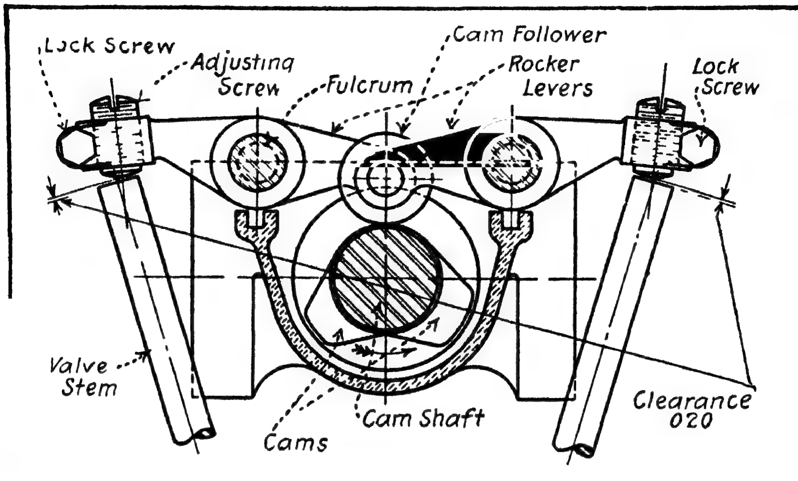


Fig. 268.—Diagram Showing Proper Clearance to Allow Between Adjusting Screw and Valve Stems in Hall-Scott Engines.

designing it is not necessary to figure on maintaining certain absolute center distances between the crankshaft and camshaft sprockets, as would be the case if conventional forms of gearing were used. On some forms of motor employing gears, three and even more members are needed to turn the camshaft. With a chain drive but two sprockets are necessary, the chain forming a flexible connection which permits the driving and driven

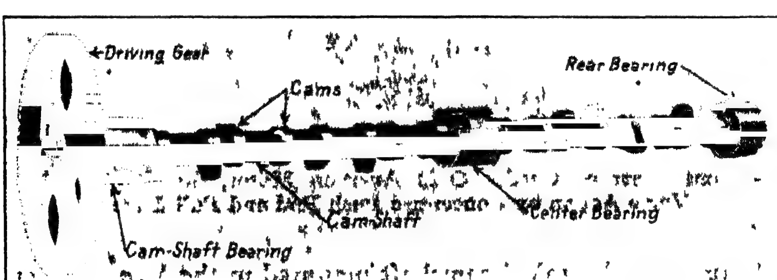


Fig. 269.—Camshaft of Eight-Cylinder "Vee" Airplane Motor Designed for Crankcase Mounting has Cams Forged Integrally. Note Split Camshaft Bearings and Method of Gear Retention by Bolting to a Flange on Camshaft.

members to be placed at any distance apart that the exigencies of the design demand. When chains are used it is advised that some means for compensating chain slack be provided, or the valve timing will lag when chains are worn. Many combination drives may be worked out with chains

that would not be possible with other forms of gearing. Direct gear drive is favored at the present time by airplane engine designers because they are the most certain and positive means, even when a number of gears must be used as intermediate drive members. With overhead camshafts, bevel gears work out very well in practice, as in the Liberty, Hispano-Suiza, Curtiss and Packard motors and others of that type. The camshafts are parallel with the crankshaft and a vertical shaft with bevel gears at each end is used to transmit motion from one horizontal shaft to the other.

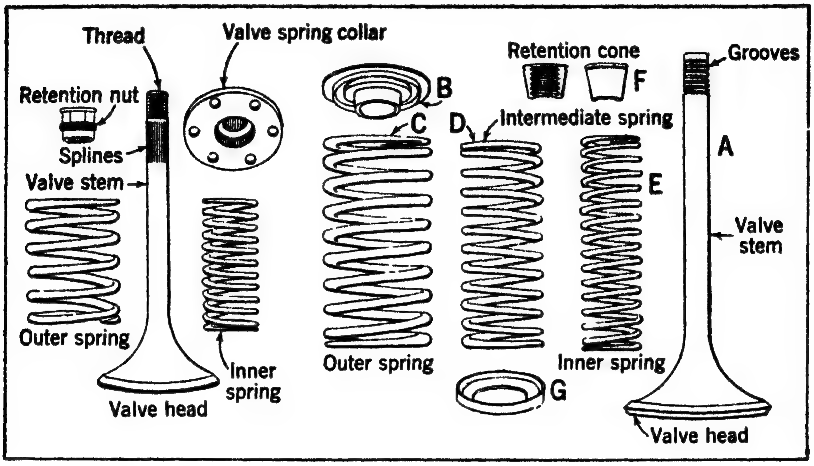


Fig. 270.—Double and Triple Valve Spring Arrangement and Method of Valve Spring Retention Used on Aviation Engines.

Valve Springs.—Another consideration of importance is the use of proper valve-springs. Particular care was needed with those of now obsolete automatic valves. The spring must be weak enough to allow the valve to open when the suction is light, and must be of sufficient strength to close it in time at high speeds. It should be made as large as possible in diameter and with a large number of convolutions, in order that fatigue of the metal be obviated, and it is imperative that all springs be of the same strength when used on a multiple-cylinder engine. Practically all valves now used to control the gas flow in airplane engines are mechanically operated as the automatic inlet found on early Anzani engines is mechanically operated in latest types and all engineers favor the positive type. On the exhaust valve the spring must be strong enough so that the valve will not be sucked in on the inlet stroke. It should be borne in mind that if the spring is too strong a strain will be imposed on the valve-operating mechanism, and a hammering action produced which may cause deformation of the valve-seat. Only pressure enough to insure that the operating mechanism will follow the cam is required. It is common automotive practice to make the inlet and exhaust valve-springs of the same tension when the valves are of the same size and both mechanically operated. This is done to simplify manufacture and not because it is necessary for the inlet valve-spring to be as strong as the other. Valve-springs of the helical coil type are generally

used, though torsion or "scissors" springs and laminated or single-leaf springs are also utilized in special applications. Sometimes volute springs are used instead of the round section helical coil springs, these being made of rectangular section steel. Two or more springs are used on each valve in some valve-in-the-head types; a spring of small pitch diameter inside the regular valve-spring and concentric with it. Its function is to keep the valve from falling into the cylinder in event of breakage of the main spring in some cases, and to provide a stronger return action in others.

The illustration at Fig. 270 shows typical valve and spring assemblies. That at the left of the cut is the old valve used on Lorraine (French) engines. Two springs were used, one inside the other. The valve-spring retaining collar was provided with splines fitting corresponding members on the valve stem. The assembly was held together by a threaded nut fitting the end of the valve stem. The new type of valve shown at Fig. 270 at the right of the cut has the method of retention commonly used on modern aviation engines. Three springs surround the valve stem A, that at E being inside, then D outside of that and inside of and concentric with the largest spring C. The spring retaining collar B has a conical depression in its top face in which the split cone members F fit. These pieces fit the grooves at the top of the valve stem and as they are kept pressed against the stem because they fit the taper cup in the collar by the spring pressure, a very positive method of retention is obtained, withal, one easy to dismantle. The split cone is easily released from the valve stem by compressing the springs with the retaining collar, a special tool of simple construction expediting the process.

Valve Spring Surge.—A study of the influence of vibration on valve-springs made by W. T. Donkin and H. H. Clark of the Cleveland Wire Spring Company and read before the S. A. E. reveals two possible ways in which a valve-spring can vibrate: First, when the frequency of the forced vibration is low, the force tending to vibrate the spring is small, due to the comparatively small centrifugal effect of the valve-operating cam, and, as the time between applications of the force is long enough to permit the vibrations set up in the spring to die out in the interval the spring vibrates as a part of the entire vibrating system. Second, if the forced vibration is of a rather high frequency and is of such a value that it is an arithmetical factor of the natural frequency of the spring, a condition of resonance may be set up whereby the spring vibrates in itself at its own frequency. An analysis of surge effected with the aid of a super-speed motion-picture film and the vibroscope, which were used in conjunction with a valve-gear testing machine, reveals that, when the spring is vibrating in its own period, at a given instant, certain adjacent coils may be spread apart beyond their normal position for an equivalent static compression and that, an instant later, the condition of these coils may be exactly reversed so far as deflection is concerned. The abnormal compression of the coils at one instant and abnormal opening of the same coils at the very next instant cause the existence of a stress and a stress range that are much greater than those calculated by conventional formulas. Furthermore, the stress range is passed through in a time equivalent to that required for the completion of

one wave of the natural period of the spring. This time factor, combined with the stress conditions existing, tends to accelerate greatly the fatigue of the valve-spring. Accordingly, the condition described is a prolific cause of valve-spring failure.

From the hypothesis of the nature of valve-spring surge, the obvious inference is that it is good practice to so design valve-springs as to obtain the highest possible natural frequency. This has been found to be true, within limits. Other factors, such as load and stress, being equal, a high-frequency spring is desirable since its wave-length is so high, compared with the wave-length of the disturbing force, that the spring will not vibrate in itself at the lower camshaft-speeds. Another, and true, inference that can be drawn from the hypothesis is that surge, because of its nature, cannot be eliminated except by recourse to some sort of friction damper which might prove expensive and annoying. It must be expected that, in the present state of the spring art, a spring always will vibrate when the frequency of the disturbing force is an integral part of the frequency of the spring, provided the frequency of the disturbing force is high enough to cause the spring to vibrate of itself. This proviso suggests a possible solution of the problem. Why not so design valve-springs that their frequency will be so high that they will not tend to vibrate until they have reached forced vibration or camshaft-speeds in excess of those encountered in customary practice?

Data relative to the characteristics of two actual springs are given in Table below. The spring whose characteristics are given in the column headed "Original" was found to have noisy vibrational periods when running in the engine, and, moreover, some cases of breakage in service were reported notwithstanding the low static-value of stress as shown in the table. It was proposed to supplant this spring by one whose characteristics are given in the last column of the table, and in which an endeavour was made to hold to the same approximate valve loads but to increase the natural frequency of the spring.

CHARACTERISTICS OF AN ORIGINAL SPRING THAT VIBRATED NOISILY AND A REDESIGNED SPRING IN WHICH THE TROUBLE WAS ELIMINATED

Item	Original Spring	Redesigned Spring
Mean or Pitch Diameter, in.	1.102	0.878
Free Length	3 1/32	2 23/32
Total Number of Coils	10 ½	9
Gauge of Wire, W. & M. No.	9	10 ½
Load, with Valve Open at 2 3/32 in., lb.	48	53
Load, with Valve Closed at 2 11/32 in. lb.	29	29
Stress, with Valve Open, lb. per sq. in.	41,200	57,000
Stress, with Valve Closed, lb. per sq. in.	25,300	31,200
Stress Range, lb. per sq. in.	15,900	25,800
Rate, lb. per in.	59.0	77.3
Weight of the Active Mass, lb.	0.1435	0.0700
Frequency, free vibrations per min.	10,750	17,600

It may be of interest to mention briefly the way in which this higher

frequency of the spring is obtained by design. First, in this case, the pitch diameter of the spring is decreased considerably, which tends to increase the rate of the spring. But to obtain the correct valve-loading and, at the same time, a reasonable free length, it is important that the rate be not increased excessively, hence the diameter of the wire is reduced somewhat, but the net result is a considerable increase in the rate. It should be noted that the number of active coils also affects the rate and that this number is varied to vary the rate. The weight of the active mass in the spring is decreased by reducing the diameter of the wire, the pitch diameter, and the number of active coils. The increase of rate together with the decrease of mass affects a substantial increase in the natural frequency of the spring in accordance with Ricardo's formula.

From the foregoing discussion it can be seen that the effect of surge upon a valve-spring is to bring about the following conditions:

(1) Existence of a maximum stress of a higher value than that calculated by the conventional formula

(2) Occurrence of a higher stress-range in the end coils than that calculated on a static basis

(3) A state whereby the stress range is traversed with great rapidity.

All investigations along the line of metal fatigue tend to show that fatigue life is a function of the maximum stress existing, the stress range through which the material is worked, and the total number of oscillations to which the material is subjected. Accordingly, the abnormal stress conditions and the abnormal rapidity with which the stress range is crossed, due to the surge, tend to accelerate greatly the fatigue of the spring.

Four Major Contentions Advanced.—These notes suffice for Messrs Donkin and Clark to advance four major contentions relative to the general subject of valve-spring surge, and they can be summarized as follows:

(1) Valve-spring surge is a product of a resonant condition between the rate of the forced vibrations, due to the camshaft, and the natural frequency of the spring

(2) Surge cannot be eliminated in the present type of spring, as the tendency to vibrate is an inherent characteristic of a spring. A solution is offered, however, by designing springs for high frequency so that they will not vibrate until they are subjected to rates of forced vibration in excess of speeds above the range of normal, maximum camshaft speeds

(3) Surge is affected by the design of the valve gear. A tendency to magnify spring surge may exist if one or more parts of the valve gear are also in resonance with the spring

(4) Surge is a prolific source of valve-spring failure, due to the stress conditions and the rapidity of the stress cycle set up by the surge.

Packard Multiple Cluster Valve-Springs.—The valve-springs of Packard aviation engines are worthy of special note. These are of the multiple cluster type and consist of a group of small diameter piano wire springs arranged in a planetary fashion around the valve stem. In the model 1,500 engine, seven of these springs and, in the model 2,500 engine, ten springs are used with each valve. The individual springs are located over tubular guides that are welded to a lower fixed washer; the upper ends of the

springs engage in an annular groove formed in the movable spring washer. Several advantages accrue from this construction, which may justifiably be termed indestructible. The most important point, perhaps, is the least obvious, namely, that which relates to the natural period of vibration of the small springs. Other advantages result from the increased factor of safety in numbers, since any valve will continue to function even though several of the springs may be broken. Furthermore, the reciprocating weight, represented by the upper washer and one-half the weight of the springs, is reduced, as compared with the conventional construction, and, finally, the physical properties of the small gauge piano wire are generally superior to those of springs heat-treated after forming.

Valve-spring failures have always been prevalent to a certain extent in aircraft engines; and these failures at times lead to disastrous results with overhead-valve engines for the valve may drop into the combustion-chamber and, consequently, wreck the piston and the combustion-chamber head. For some time, these failures were regarded as not being preventable, the cause being attributed to fatigue and to minute imperfections in the material. It is clearly proved that the basic seat of the trouble lies in a resonance effect between the natural vibrations of the spring and the forced oscillations of the engine. These latter oscillations are brought about by the firing impulses.

The Packard engineers had noted that, in very high-speed six-cylinder engines, valve-spring breakages were frequently encountered at speeds in excess of 4,000 r.p.m.; in twelve-cylinder engines the limiting speed appeared to be above 2,000 r.p.m.; and in some eighteen-cylinder engines frequent valve-spring failures occurred at very moderate speeds, certainly not exceeding 1,600 r.p.m. Naturally, the valve-springs in each case were of somewhat different design from the others but the variations were not of sufficient magnitude to refute the statement that the critical engine speed at which valve-spring failures assumed alarming proportions was inversely proportional to the number of cylinders or of the firing impulses. The fact that the small springs have been immune from failure, after a great many prolonged tests with high-speed engines, goes a long way toward substantiating the claim that valve-spring breakage in the past has been brought about by synchronized vibrations.

Springless Valves.—Springless valves are a late development on French racing car engines, and it is possible that the positively-operated types will be introduced on aviation engines also although the actuating and return mechanism necessary is heavier than that required for spring returned valves. Two makes of positively-actuated valves are shown at Fig. 271. The positive-valve motor differs from the conventional form by having no necessity for valve-springs, as a cam not only assures the opening of the valve, but also causes it to return to the valve-seat. In this respect it is much like the sleeve valve motor, where the uncovering of the ports is absolutely positive. The cars having motors equipped with these valves were a success in long-distance auto races. Claims made for this type of valve mechanism include the possibility of a higher number of revolutions and consequently greater engine power. With the spring-controlled, single-cam operated valve a point is reached where the spring is not capable of

returning the valve to its seat before the cam has again begun its opening movement. It is possible to extend the limits considerably by using a light valve with a strong spring or by using multiple springs but the valve still remains a limiting factor in the speed of the motor. A part sectional view through a cylinder of an engine designed by G. Michaux is shown at Fig. 271 A. There are two valves per cylinder, inclined at about ten degrees from the vertical. The valve-stems are of large diameter, as owing to positive control, there is no necessity of lightening this part in an unusual degree. A single overhead camshaft has eight pairs of cams such as shown in detail at B. For each valve there is a three-armed rocker, one arm of

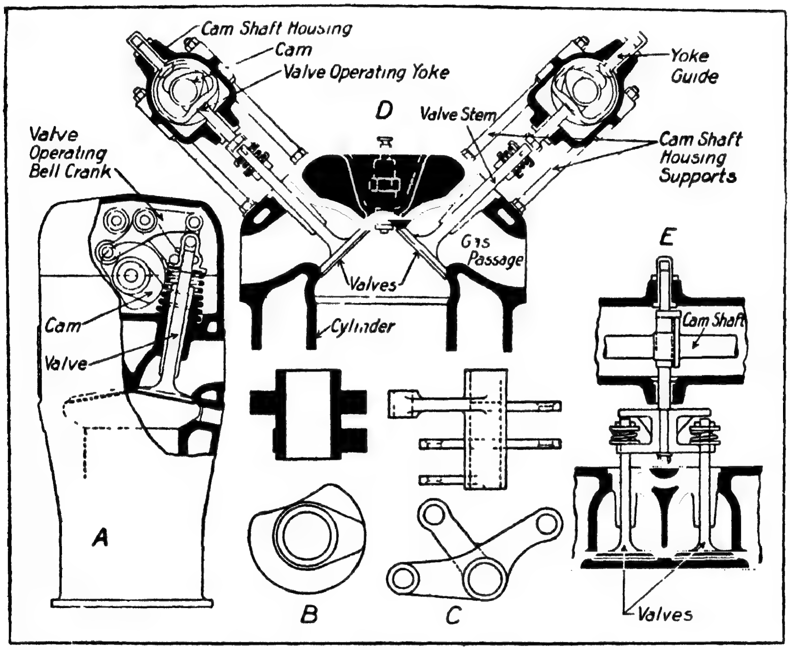


Fig. 271.—Two Methods of Operating Automotive Engine Valves by Positive Cam Mechanism which Closes as Well as Opens Them.

which is connected to the stem of the valve and the two others are in contact respectively with the opening and closing cams. The connection to the end of the valve-stem is made by a short connecting link, which is screwed on to the end of the valve-stem and locked in position. This allows some adjustment to be made between the valves and the actuating rocker. It will be evident that one cam and one rocker arm produce the opening of the valve and that the corresponding rocker arm and cam result in the closing of the valve. If the opening cam has the usual convex profile, the closing cam has a correspondingly concave profile. It will be noticed that a light valve-spring is shown in drawing. This is provided to give a final seating to its valve after it has been closed by the cam. This is not absolutely necessary, as an engine has been run successfully without these springs though it is difficult to understand how valve stem expansion is

compensated for unless there is some clearance in the valve actuating mechanism. The whole mechanism is contained within an overhead aluminum cover.

The positive-valve system used on the De Lage motor is shown at Fig. 271 D. In this the valves are actuated as shown in sectional views D and E. The valve system is unique in that four valves are provided per cylinder, two for exhaust and two for intake. The valves are mounted side by side, as shown at E, so the double actuator member may be operated by a single set of cams. The valve-operating member consists of a yoke having guide

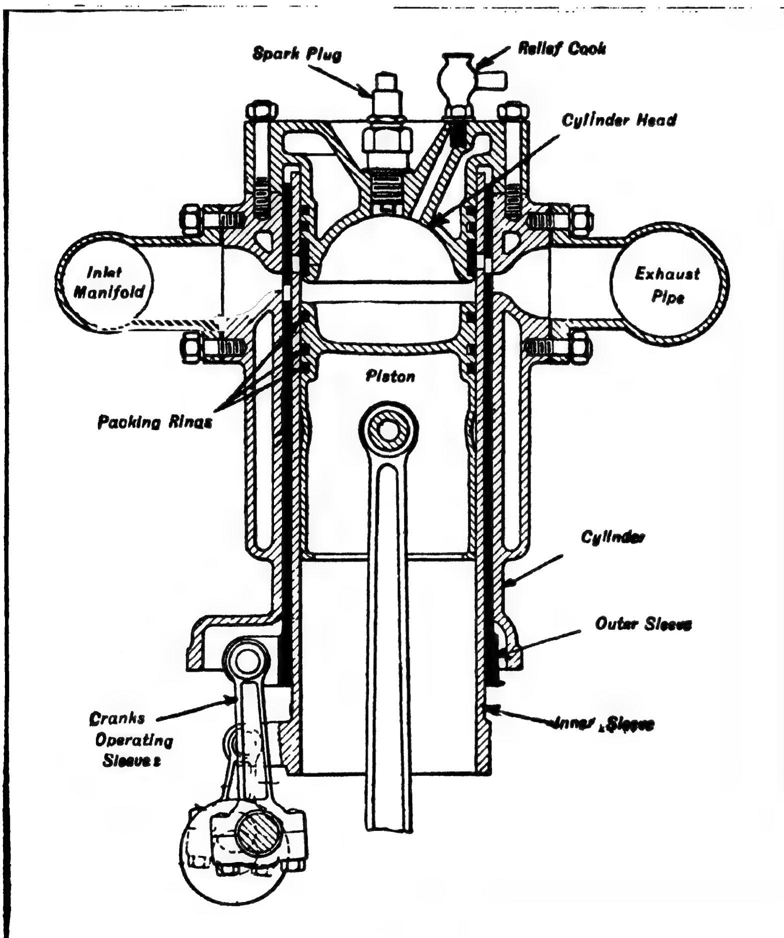


Fig. 272.—Section Through Cylinder of Knight Sleeve Motor Showing Important Parts of Valve Mechanism.

bars at the top and bottom. The actuating cam works inside of this yoke. The usual form of cam acts on the lower portion of the yoke to open the valve, while the concave cam acts on the upper part to close the valves. In this design provision is made for expansion of the valve-stems due to

heat, and these are not positively connected to the actuating member. As shown at E, the valves are held against the seat by short coil springs at the upper end of the stem These are very stiff and are only intended to provide for expansion A slight space is left between the top of the valve-stem and the portion of the operating member that bears against them when

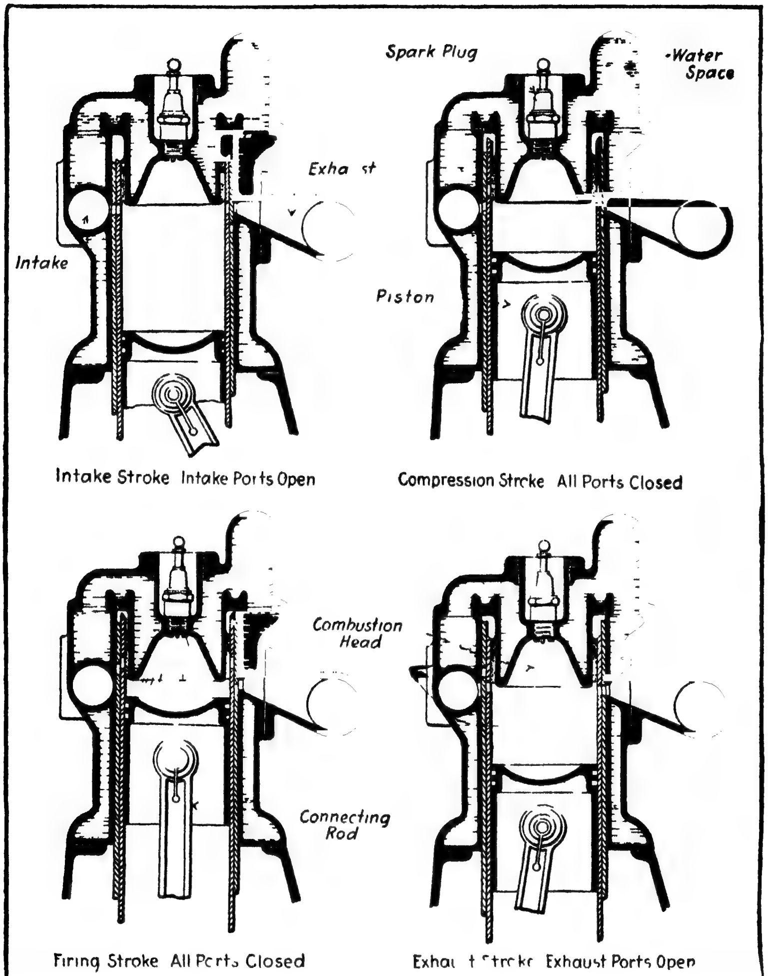


Fig. 273.—Diagrams Showing Positions of Sleeve Valves, at Different Points of Knight Engine Cycle.

the regular profile cam exerts its pressure on the bottom of the valve-operating mechanism Another novelty in this motor design is that the camshafts and the valve-operating members are carried in casing attached above the motor by housing supports in the form of small steel pillars The overhead camshafts are operated by means of bevel gearing.

Knight Sleeve Valve Motor.—The sectional view through the cylinder at Fig. 272 shows the Knight sliding sleeves and their actuating means very clearly. The diagrams at Fig. 273 show graphically the sleeve movements and their relation to the crankshaft and piston travel. The action may be summed up as follows: The inlet port begins to open when the lower edge of the opening of the outside sleeve which is moving down

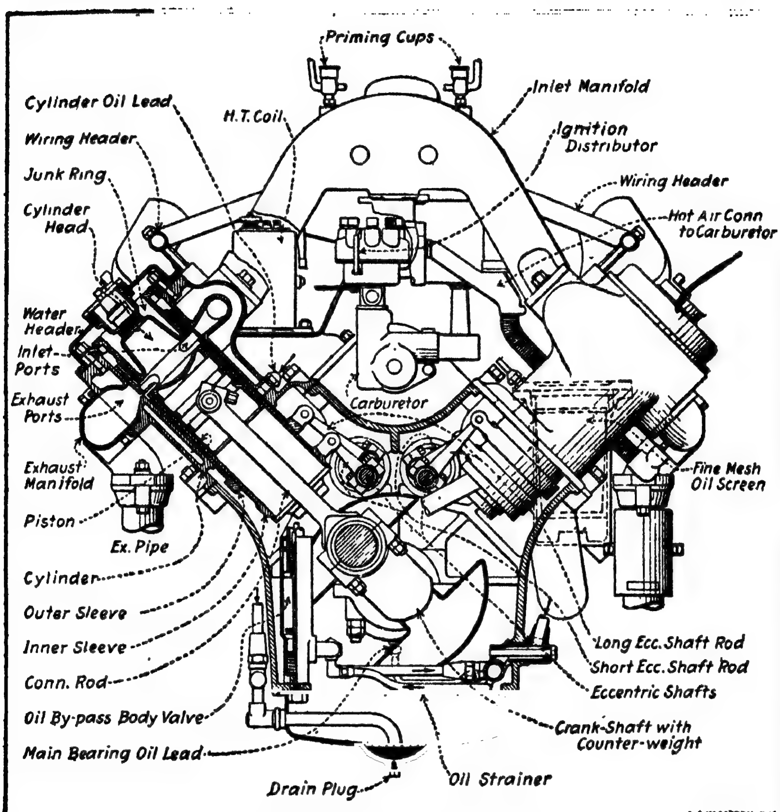


Fig. 274.—Cross Sectional View of Knight Type Eight-Cylinder "Vee" Engine for Automotive Applications.

passes the top of the slot in the inner member also moving downwardly. The inlet port is closed when the lower edge of the slot in the inner sleeve which is moving up passes the top edge of the port in the outer sleeve which is also moving toward the top of the cylinder. The inlet opening extends over 200 degrees of crank motion. The exhaust port is uncovered slightly when the lower edge of the port in the inner sleeve which is moving down passes the lower edge of the portion of the cylinder head which protrudes in the cylinder. When the top of the port in the outer sleeve traveling toward the bottom of the cylinder passes the lower edge of the slot in the cylinder wall the exhaust passage is closed. The exhaust opening extends over a period corresponding to about 240 degrees of

crank motion. The Knight motor has been applied to aircraft to the writer's knowledge only by the Panhard firm. An eight-cylinder Vee design for automobile use that might be useful in that connection if lightened is shown at Fig. 274. The main object is to show that the Knight valve action is the only other besides the mushroom or poppet valve that has been applied commercially to high-speed gasoline-engines, though many types of rotary and single sleeve valves exist.

Burt-McCollum Sleeve Valve.—The only sleeve valve engine that has stood the test of time besides the Knight motor employs the Burt-McCollum single sleeve valve and it was first introduced in 1911. It was first applied in motor cars by Argylls, Ltd. of Alexandria, Scotland. Sometime ago, it was announced that the sole manufacturing rights for the United States were held by Continental Motors and engines are under test for automobiles and airplanes.

Burt's original patent was filed on Aug. 6, 1909, and was accepted on Aug. 8, 1910. A four-year extension was granted last year, so that the date of expiration in Great Britain is Aug. 5, 1929. Application for an equivalent United States patent was filed on Aug. 3, 1910, and was granted on July 22, 1919, thus giving an expiration date of Dec. 3, 1936. Perhaps the broadest claim contained in the United States patent is that which reads:

> A mechanism for the purpose specified comprising a main cylinder having intake and outlet ports near its head, a piston-enclosing cylinder movably fitting within said main-cylinder to act as a valve and having intake and outlet ports, and movable, for bringing the respective ports into alternating registering relation therewith, a piston reciprocating within said enclosing cylinder, and means for imparting synchronous longitudinally reciprocating and oscillating movement to said piston-enclosing cylinder, the range of longitudinal movement being less than the piston movement and greater than the longitudinal dimension of the ports.

J. H. K. McCollum's patents, which were assigned to Argylls, Ltd., and others, consist of a sleeve outside the main cylinder, instead of between the cylinder and the piston, and two operating-mechanisms for imparting the reciprocating and oscillating movement to the sleeve.

The following description is taken from a paper by W. A. Frederick that appeared in the *S. A. E. Journal* for May, 1927.

Advantages of the Single-Sleeve-Valve Engine.—The chief advantages of a single-sleeve-valve engine are:—(a) sustained operating-efficiency, (b) good power output, and (c) silence in operation. A 1,000-hour test-run under full load at 2,000 r.p.m. was recently made on a six-cylinder 2⅞ by 4¼ inch engine. During the first 100 hours, the power-output gradually built up to 44 brake horsepower, after which it remained constant until the completion of the test. Measurement of sleeve driving-gear back-lash was made by an extended arm attached to the sleeveshaft and, although a maximum increase of 0.017 inch on the pitch-line was recorded, the gears ran as quietly as at the beginning of the test. On dismantling the engine, the maximum wear on the piston skirt was found to be 0.001 inch, while the wear on the outside diameter of the sleeve was undiscernible.

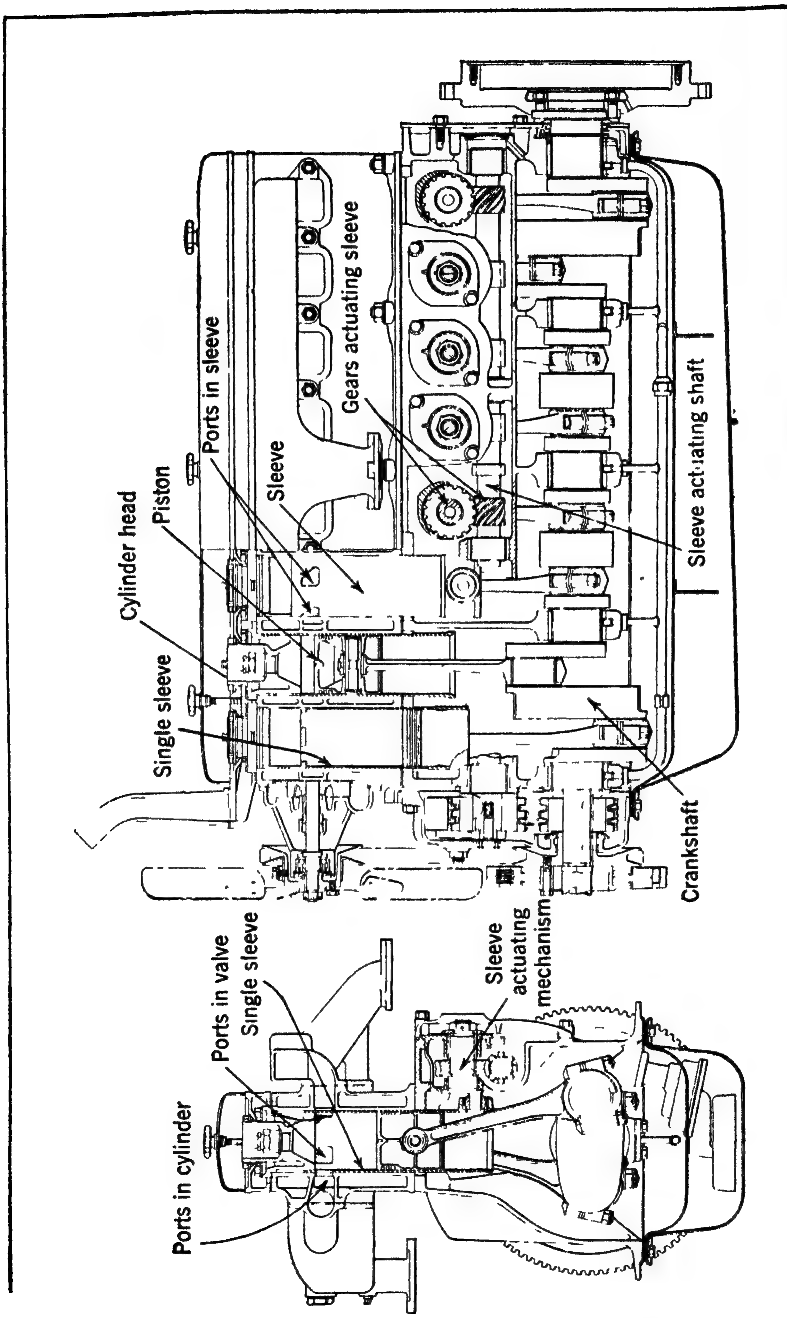


Fig. 274A.—Sectional View of Automobile Type Single Sleeve Valve Motor.

The adoption of a sleeve valve is said to obviate such things as the "grinding-in" of valves, the ingress of unwanted air through worn valve-guides, the adjustment of clearance, the breakage of valve-springs, and frequent decarbonization. The rapid opening of the ports, the type of port-opening obtained, positive timing, unobstructed intake-passages, and increased compression-ratio all contribute to good power-output. Silence in operation is achieved through avoiding the hammer-and-anvil blows of a poppet-valve and the fact that the valve-actuating mechanism does not extend outside the engine body. The sleeve-valve engine can not only be made to run quietly, but remains quiet.

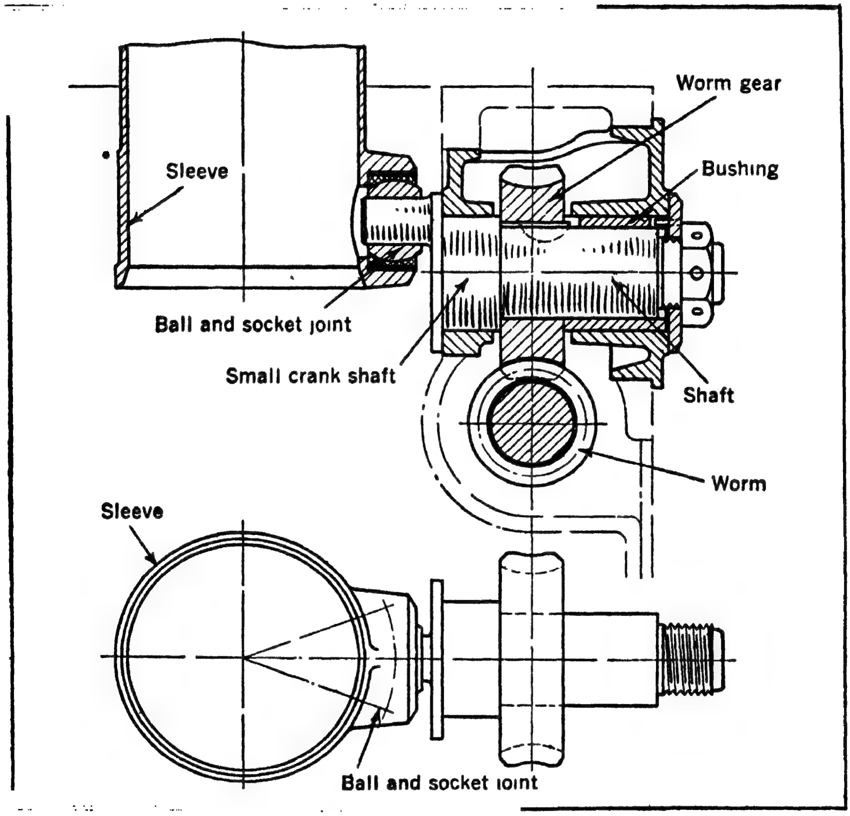


Fig. 274B.—Method of Actuating Single Sleeve Valve by Gear Actuated Crank and Ball and Socket Joint.

The weight of a single-sleeve-valve engine compares favorably with that of a poppet-valve engine in the automobile types, though it will be slightly heavier in the forms intended for airplane use. Sectional views of the Continental eight-in-line automobile motor are shown at Fig. 274 A.

Referring to the cross-sectional arrangement shown in Fig. 274 A, it will be seen that the crankshaft, connecting-rod and piston are of conventional design and not necessarily different from those used in a poppet-valve type of engine. The essential difference lies in the substituting of a single

valve of cylindrical form for the usual poppet-valves.

The cylinder is open-ended and has port-openings cut on its circumference immediately below the bottom edge of the cylinder-head or stationary piston. Intake and exhaust-ports are on opposite sides of the cylinder-block so that separate manifolds are employed. The tubular steel sleeve is interposed between the piston and cylinder having ports machined therein to register with corresponding members in the cylinders. The size and the shape of the ports are determined by the area and the valve-timing required. In common with those of the poppet-valve engine, these can only be settled by experience, each particular type of engine being considered according to the performance required of it.

Motion of Single Sleeve.—The characteristic twisting movement of the sleeve has many inherent advantages. Although primarily conceived for the purpose of obtaining the proper sequence of valve operation, when using one sleeve instead of two, it has subsequently proved to be a decided benefit in other respects. It is a natural lubricating motion, the oil being rolled evenly over the entire sleeve-surface, and not localized and sheared, as in the case of a sleeve or piston having a reciprocating motion only. Oil grooves are not necessary on the surface of a single-sleeve valve.

The movement of the sleeve approaches harmonic motion, and does not call for the sudden reversal of the direction of travel with its attendant inertia-loading, as does a sleeve with purely reciprocating motion.

Again, the twist dissipates the heat lost to the sleeve wall over a larger area, giving more even temperatures and therefore reducing the distortion to the minimum. During the compression and power strokes, when the sleeve is subjected to the greatest pressure, it is moving with the piston, and the sleeve-ports are protected between the water-cooled surfaces of the cylinder and the cylinder head.

It has not been found necessary to fit a sealing-ring to the cylinder head, such as that used in the double-sleeve-valve engine. This is due, no doubt, to the baffle effect caused by the twisting movement that smoothes out minute surface irregularities, and to the fact that there is a sealing surface at both sides of the sleeve wall during compression.

Because of the shape of the combustion-chamber, a comparatively higher compression-ratio can be adopted without fear of detonation, five to one being generally used on automotive engines.

The fact that the piston and the sleeve move in the same direction at different speeds during the pressure strokes results in a considerable reduction in the piston rubbing-speed, as compared with that of a conventional poppet-valve engine, and so reduces the wear of the piston. The wear of the sleeve surface is practically negligible. It will be noticed that the sleeveshaft crank is near the 90-degree position at the time of the closing of the intake and the opening of the exhaust, so that the sleeve moves with the maximum velocity in a practically vertical direction, and therefore gives the desirable quick opening and closing of the valve-ports.

The sleeves are usually made of cylinder cast iron, cast in a rotating mould, although ordinary sand-castings are entirely satisfactory, if carefully made. The thickness of the wall of the sleeve is governed by what the

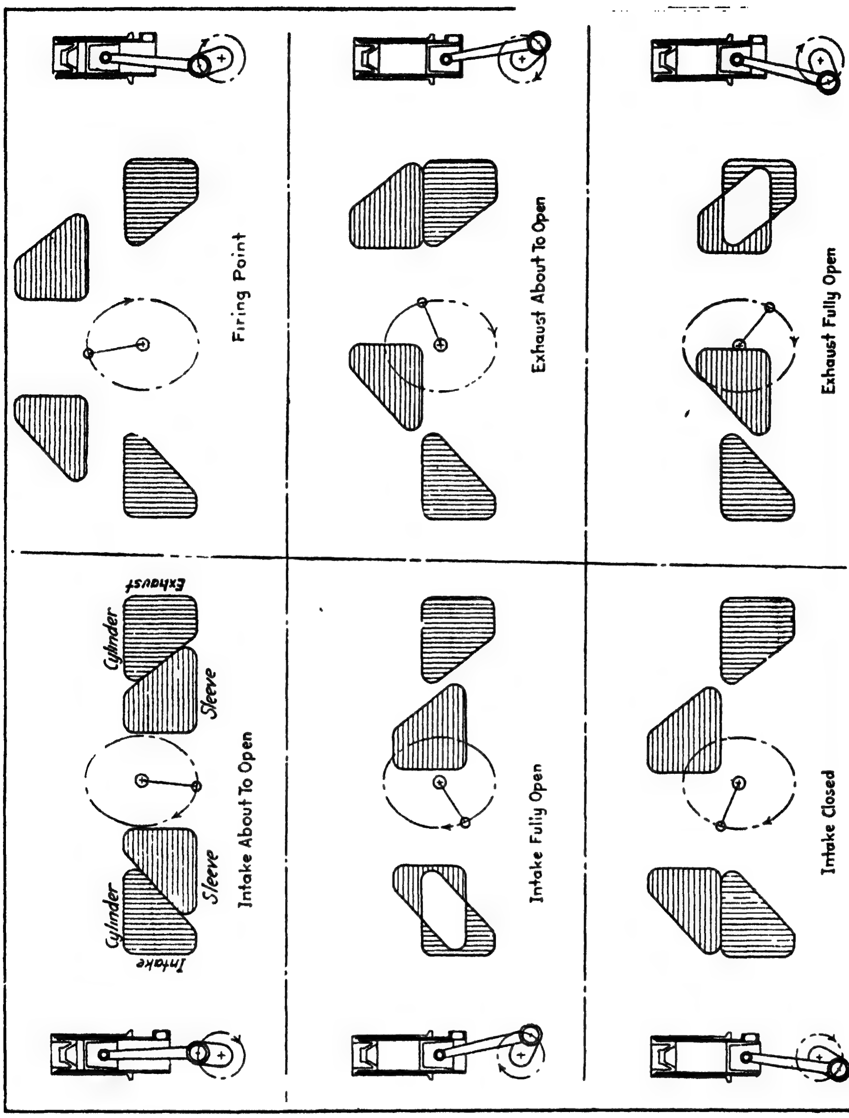


Fig. 274C.—Valve Timing Diagram Showing Sleeve Port Openings and Corresponding Piston Positions.

machine-shop can handle without fear of breakage. This will be found ample for strength under working conditions. In practice, the thickness of the wall ranges from $\frac{5}{64}$ inch for the $2\frac{3}{4}$ inch bore to $\frac{9}{64}$ inch for the five inch bore.

Steel sleeves are sometimes used when high engine speed is desired. It is usual to manufacture them from seamless steel-tubing, the boss for actuating being formed by extruding operations. A sleeve of average diameter is fitted to the cylinder-bore with a tight 0.003 inch and a slack 0.002 inch feeler.

Sleeve Drive Mechanism.—The ball-and-socket connection shown at Fig. 274 B has evolved from a self-aligning ball bearing to the present sphere-zone, having a pressure die-cast babbitt-socket cast in position. It has been found that the fit of the ball in the socket can be varied by the pressure applied. The diameter of the sphere-zone is generally made 0.35 D, while the sleeve-shaft crankpin approximates 0.19 D, where D equals the outside diameter of the sleeve. Originally, the ball and the socket were made detachable, but this reduced the bearing area due to flats milled on the ball to allow assembling. The design of the mechanism for actuating the sleeve is an interesting problem and many schemes have been tried out from time to time. All things considered, however, the gear type of drive in various forms has proved the most satisfactory.

Single Sleeve Timing Diagram.—The port-cycle diagram, Fig. 274 C, shows one complete cycle of sleeve movement, the relative positions of the piston being indicated diagrammatically at the side of the drawing. At the beginning of the intake stroke, the sleeve is at its bottom center, and all ports are closed (upper left view). As the piston descends, the sleeve moves around and up the lower left-hand portion of the travel ellipse, the intake-ports in the sleeve uncovering the intake-ports in the cylinder (central left view). Intake closing occurs when the bottom straight-edge of the sleeve-port coincides with the top straight-edge of the fixed port in the cylinder, lower left view. As the piston turns on the compression stroke, the sleeve continues to travel upward, reaching its top center at the same time as the piston (upper right view). During the power stroke, the sleeve moves over and down the top right-hand portion of the travel ellipse. Exhaust opening takes place when the bottom edge of the sleeve exhaust-port meets the top edge of the port in the cylinder (central right view), the sleeve moving downward as the piston moves upward on the exhaust stroke. In the lower right view, the maximum exhaust-opening occurs, the sleeve traveling on the lower right-hand portion of the ellipse until the flank edges of the ports coincide, closing the ports and completing the cycle.

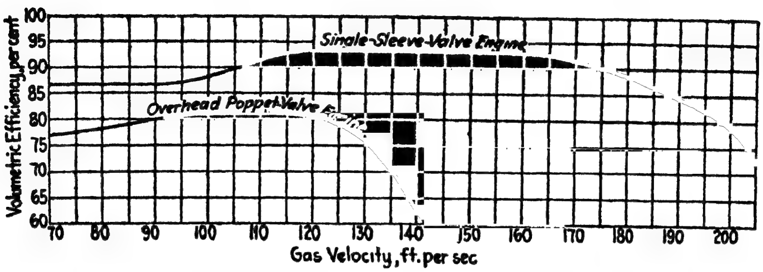
AVERAGE TIMING-PRACTICE OF SINGLE-SLEEVE-VALVE ENGINES

Type of Engine	Intake Opens	Intake Closes	Exhaust Opens	Exhaust Closes
Low-Speed	10 Deg. Late	15 Deg. Late	45 Deg. Early	10 Deg. Late
Medium-Speed	10 Deg. Late	20 Deg. Late	45 Deg. Early	10 Deg. Late
High-Speed	5 Deg. Late	30 Deg. Late	60 Deg. Early	15 Deg. Late
Timing Tolerances Allowed on Production	2 Deg. Early to 5 Deg. Late	4 Deg. Early to 4 Deg. Late	4 Deg. Early to 4 Deg. Late	5 Deg. Early to 2 Deg. Late

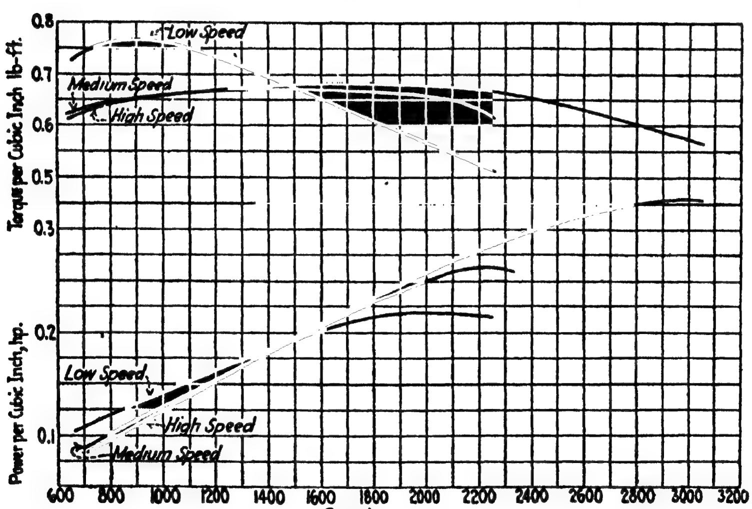
Fig. 274 D shows the power output per cubic inch of three engines having valve-timing as given in Table above. The engines from which the

curves were taken differed in some respects, so that an absolutely true comparison of the effect of timing only is not presented.

Detachable Head Construction.—A detachable head for each cylinder has the advantage of obviating the use of a large casting and gasket. Any cylinder may be examined without disturbing the joints of the others. Explosion balance is assured, as the combustion space is completely



—Volumetric-Efficiency Graph of a Poppet-Valve Engine and of a Sleeve-Valve Engine of Equal Capacity



—Curves of Power Output and Valve-Timing of Single-Sleeve Valve Engines

Fig. 274D.—Curves Showing Characteristics of Single Sleeve Valve Engines.

machined, and the sparkplug is ideally placed and effectively water cooled. Cast iron is usually employed, although aluminum is satisfactory and is used where lightness is of first importance.

Many shapes of combustion-chamber have been tried, but, although the hemispherical type is theoretically ideal, it has not proved in practice to be as good as the cone-frustrum type.

The head is made a light push-fit in the sleeve and is secured by four cap-screws, an extra-thin gasket being interposed between the head-flange and the cylinder. It should be noted that the head-joint is not subjected

to direct explosion-pressures.

In common with that of other sleeve-valve engines, its power output improves as the carbon builds up around the head. A standard sparkplug is used; the long-reach sparkplug and extension, characteristic of the early models, has been discarded due to adoption of the cone-frustrum type of cylinder head.

The number of ports incorporated in the design of engines of different types ranges from two intake and two exhaust to four intake and four exhaust. For a given area, the smaller the number of ports, the greater will be the degree of filling, but the fewer the ports, the greater will be the sleeveshaft throw required; so, as in the case of many other engineering conditions, a compromise must be made. Three intake and two exhaust-ports have been found to be the best all-round combination for automobile engines. This arrangement gives a sleeveshaft throw of moderate dimensions and practicable water-cores between the ports in the cylinder, while the maximum port-opening area obtainable compares favorably with good poppet-valve practice, a condition that has proved satisfactory for average engines.

Minerva-Bournonville Rotary Valve.—Although designed in this country, the Minerva-Bournonville rotary valve engine was perfected in the experimental department of Minerva Motors, Antwerp, Belgium. While there is no record of its use in aviation engines, the suggestion that rotary valves be used for that purpose is often made and the following description, from *Automotive Industries* is given in part for the general information of the reader interested in engine design. The designer of this valve is Eugene Bournonville, a well known engineer residing in New York.

The Minerva-Bournonville engine possesses a rotary valve mounted in a bore in the head of the cylinder and making one revolution for six turns of the crankshaft. For each cylinder the valve has three pockets which assure communication between the intake and the exhaust ports on the one hand, and a port to the cylinder on the other. On compression and expansion the cylindrical portion of the rotary valve masks the passage into the cylinder. The sparkplug is mounted in the usual inclined position.

All previous attempts to perfect the rotary valve engine have failed by reason of inability to hold compression or to assure adequate lubrication. The Minerva-Bournonville overcomes this by means of what may be described as an elastic bearing. The tunnel in which the valve rotates is cylindrical on an arc of only a few degrees. Carried on the valve, in the vertical axis of the cylinder, is a cast-iron shoe surmounted by a cast-iron wedge, with a hardened steel ball between shoe and wedge. The wedge exerts pressure on the shoe, and through it on the rotary valve, by means of a stud and an adjustable coil spring placed in a horizontal passage in the valve casing. In the case of the six-cylinder engine, the valve is in two parts, the shoes are in four parts, the wedges are in two parts, there are two balls under each wedge and there are two springs pressing against the vertical wall of each wedge.

Lubrication of the valve mechanism is assured by a distributor driven off the rear end of the rotary valve shaft. This distributor sends a drop of oil at intervals of 150 revolutions of the crankshaft to four points on the

top of the wedges opposite the springs. In addition there is an oil lead to the intake manifold also giving one drop of oil per 150 revolutions.

The bore of the valve tunnel gives sufficient clearance to allow for the maximum expansion of the valve so that seizure at this point is impossible. The shoes are cut out of a tube bored to a diameter slightly larger than that of the valve, the difference being about $\frac{1}{1000}$ of the diameter. They are turned externally to assure an equal thickness over their entire width. The wedges have an angle of seven to seven and one-half degrees. The steel ball B, mounted between the wedge and the shoe, is free in a vertical direction and only works laterally; when the shoe is carried round by the valve, it communicates this turning movement to the wedge, which in its

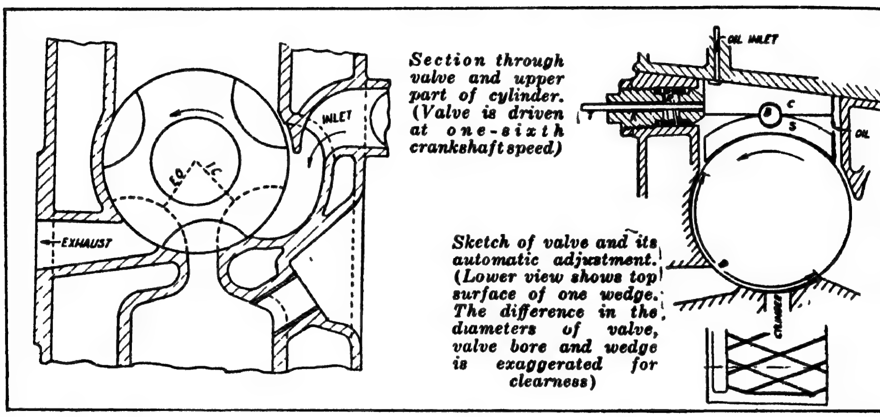


Fig. 274E.—Minerva-Bournonville Rotary Valve.

movement tends to compress the spring. The pressure of the wedge is therefore direct on the shoe. Adjustment of the mechanism is simple and makes it possible to determine externally whether there is any movement of the shoe and the wedge, which movement could be set up by dilation of the valve or by abnormal friction caused by lack of oil. After the engine has been started up, the adjusting plug A Fig. 274 E is screwed down on the spring, and by means of a short rod passing through the plug any movement of the wedge C can be detected. If the plug is screwed down until there is no movement of the rod passing through it, this is an indication that the wedge is exerting sufficient pressure on the valve to prevent it lifting under explosion. The driving effort remains constant, for it depends on the coefficient of friction of the valve and the shoe, and the pressure at the moment of explosion.

Movement of the shoe and the valve can take place only for two reasons, as follows:—(1) Dilation of the valve increases the thrust on the shoe, causing this to turn and take up a corresponding position; (2) the coefficient of friction is increased by reason of foreign matter, such as dust or an unlubricated portion of the roller, this latter tending to carry the shoe round with it, thus displacing the wedge and immediately decreasing the pressure. Under all conditions the valve operates freely, while providing sufficient pressure to assure gas-tightness. The shoes are bored to a diameter equal

to that of the valve at its maximum temperature, and the part of the cylinder D, in the drawings, has the same diameter. The difference between this and the normal diameter of the valve is slight, being not more than 1:1,000, and the valve exerts a sufficient pressure on the edges of the ports during the period of starting up to assure gas-tightness until the normal temperature has been attained. After this the valve bears equally on the entire surface. By reason of the design any required timing is possible and the biggest desirable openings are obtainable.

Valve Design and Construction.—Valve dimensions are an important detail to be considered and can be determined by several conditions, among which may be cited method of installation, operating mechanism, material employed, engine speed desired, manner of cylinder cooling and degree of lift desired. A review of various methods of valve location has shown that when the valves are placed directly in the head we can obtain the ideal cylinder form, though larger valves may be used if housed in a separate pocket, as afforded by the "T" head construction. The method of operation has much to do with the size of the valves. For example, when an automatic inlet valve was employed it was good practice to limit the lift and obtain the required area of port opening by augmenting the diameter. Because of this an inlet valve of the automatic type was usually made twenty per cent larger than one mechanically operated. When valves are actuated by cam mechanism, as is now invariably done, they are often made the same size and are interchangeable, which greatly simplifies manufacture. The relation of valve port area to cylinder bore is one that has been discussed for some time by engineers. The writer's experience would indicate that valve diameter should be as near to half the bore as possible. While the mushroom type or poppet valve has become standard and is the most widely used form at the present time, there is some difference of opinion among designers as to the materials employed and the angle of the seat. Most valves have a bevel seat, though some have been made flat seating. The flat seat valve is said to have the advantage of providing a clear opening with lesser lift, this conducing to free gas flow, but the disadvantage is present that best material and workmanship must be used in their construction to obtain satisfactory results. As it can be made very light it is particularly well adapted for use as an automatic inlet valve or as a check valve. Among other disadvantages cited is the claim that it is more susceptible to derangement, owing to the particles of foreign matter getting under the seat and it will not conduct heat away as well as a bevel seat valve head. With a bevel seat it is argued that the foreign matter would be more easily dislodged by the gas flow, and that the valve would close tighter because it is drawn positively against the bevel seat. Bevel seatings are usually at 30 or 45 degrees, the latter being most popular. The valve seat of the Curtiss D12 engine is shown at Fig. 275. The way the valve seat is chamfered or relieved is clearly shown in the diagram which may be considered typical of good practice.

Several methods of valve construction are the vogue, the most popular form for aviation engines being the one-piece type; those which are composed of a head of one material and stem of another are seldom used in

airplane engines because they are not reliable. In the built-up construction the head is usually of high nickel tungsten steel or cast iron, which metals possess good heat-resisting qualities Heads made of these materials are not likely to warp, scale, or pit, as is sometimes the case when ordinary grades of machinery steel are used. The cast-iron head construction is not popular because it is difficult to keep the head tight on the stem. There is a slight difference in expansion ratio between the head and the stem, and

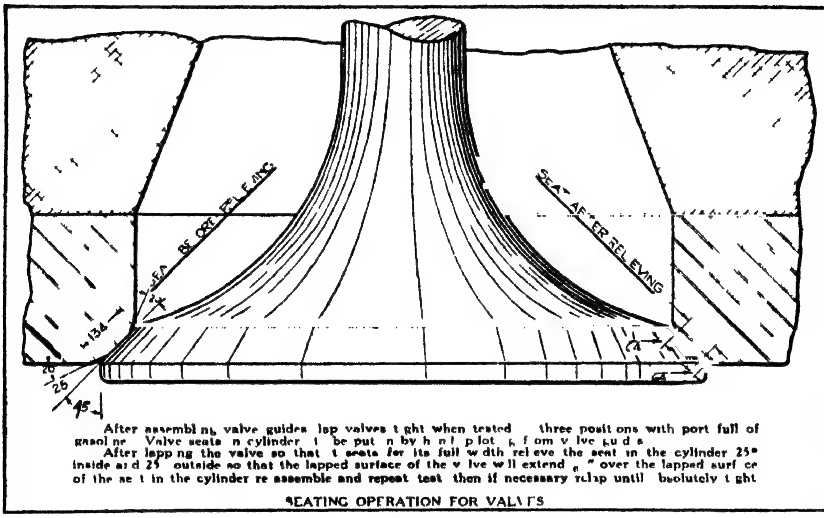


Fig. 275.—Drawing Showing Shape of Valve Seat Used on the Curtiss D12 Water-Cooled Aviation Engine.

as the stem is either screwed or riveted to the cast-iron head the constant hammering of the valve against its seat may loosen the joint As soon as the head is loose on the stem the action of the valve becomes erratic The best practice is to machine the valves from tungsten and other alloy steel forgings. This material has splendid heat-resisting qualities and will not pit or become scored easily Even the electrically welded head to stem types which are used in automobile engines are not looked upon with favor in the aviation engine. Valve-stem guides and valve-stems must be machined very accurately to insure correct action. The usual practice in automobile engines is shown at Fig. 276, where a valve-stem .375 inch nominal diameter is used.

Gas Velocity Effect on Power.—Several methods are in use as a basis for calculating gas velocities. One merely considers the mean velocity in the port, on a basis of filling or exhausting the cylinder during 180 degrees of crankshaft rotation This method is crude, as no account is taken of valve-lift, this latter factor having a considerable influence on the power output A second method considers the mean velocity through a cylindrical annulus of a diameter equal to the bore of the mouth of the port and a height equal to the maximum valve-lift, this annulus being taken as constant throughout the period of filling or exhausting The time of filling or exhausting is taken as 180 degrees of the crankshaft rotation, the cylinder

is assumed to be completely filled and the charge is assumed to be at normal temperature and pressure. This method takes no account of the mean valve-lift or of timing, both of which factors have a considerable influence on the performance. Cylinders having very small valves will usually show an improvement in the power output when an increase of the mean valve-lift is made without any change of the maximum lift. The same holds true as regards timing, very small valves with high gas-velocities generally requiring freak timing with long opening-periods to give the maximum performance. The results of some tests made with various sizes and combinations of inlet and exhaust valves by Mr. S. P. Heron are shown graphically at Fig. 277, the power indicated being that of the one test cylinder at various speeds. The highest power was obtained with four valves.

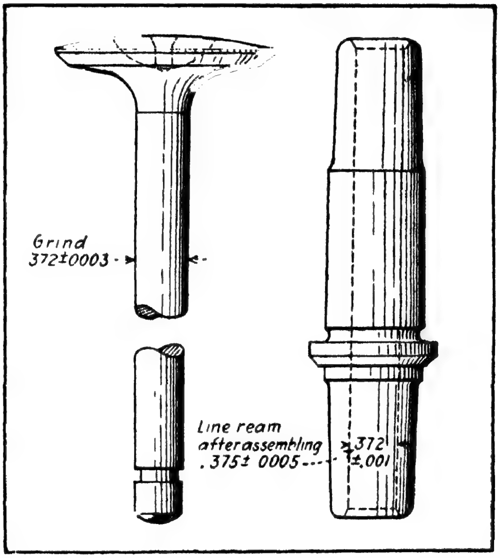


Fig. 276.—Drawing Showing Clearance Allowed Between Automotive Engine Valve Stem and Valve Stem Guide to Secure Free Action.

A third method considers the mean of the true areas in the annulus normal to the seat, the true time of opening and 100 per cent volumetric efficiency. This method is the soundest, but is too cumbersome for general use as it involves a graphical layout to determine the areas of the conical annuli from zero to the maximum lift. With this method annuli areas in excess of the net area in the throat, or the port area less the valve-stem area, are neglected. The conical annulus as a basis for the calculation of area is cumbersome, as its relation to the cylindrical annulus of equal height diminishes progressively with the lift. It is considered that a reasonable and not altogether cumbersome basis for the estimation of velocity is that of the cylindrical annulus. This is supported to a certain extent by the fact that a 45-degree seat valve, with less area in the annulus than either a flat seat or a 30-degree seat valve will, in overhead-valve cylinders, pass at least as much gas as either of the latter types at equal lift, showing that the reduction of area is balanced by the more nearly streamline-flow conditions.

As the tulip type certainly does not pass less gas than the flat-head type, it appears that the theoretical increase of area with the flat-head type of valve is of no practical moment, and that the gas does not depart from streamline-flow conditions and abruptly change its course around sharp corners to take advantage of sudden increases of area.

It is Mr. Heron's view that the all-around best valve-seat angle for overhead-valve cylinders is 45 degrees. The seat angle is considered to be most important in multiple-valve cylinders with pairs of valves in close proximity. The converging gas-streams with pairs of similar valves will meet less nearly head-on with 45-degree seats than with flat or 30-degree seats.

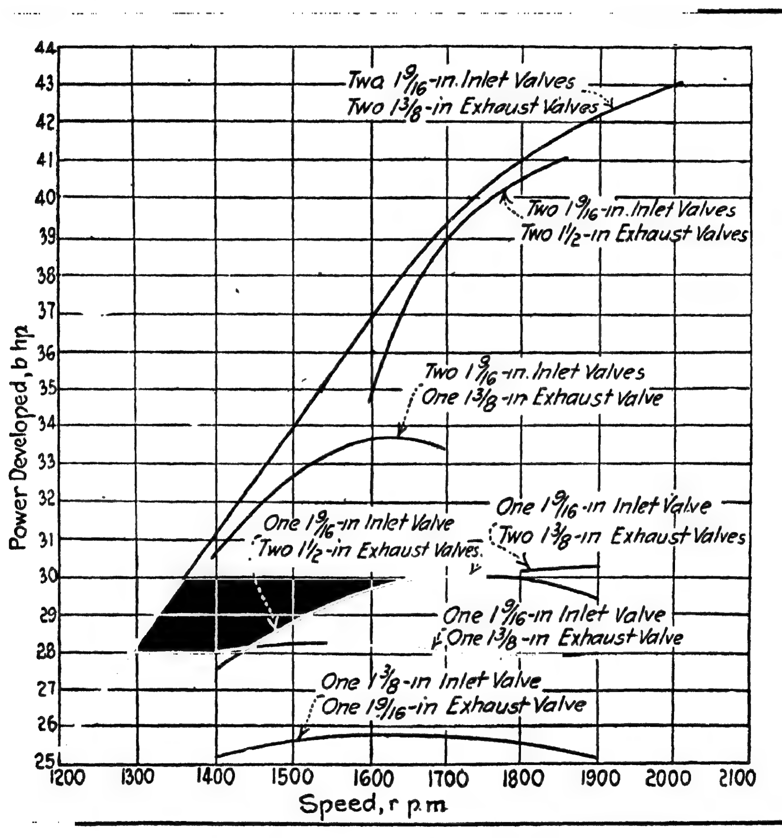


Fig. 277.—Results of Tests Made with Various Sizes and Combinations of Inlet and Exhaust Valves.

As a result of tests with an air-cooled cylinder Mr. Heron is of the opinion that exhaust and inlet valves should be of approximately equal diameter and have equal lifts and that a mean gas velocity through the valve annulus of from 140 to 160 feet per second seems to give good results in case of the particular cylinder tested.

Four Valves per Cylinder.—A great power output for the piston displacement is usually made possible by the superior volumetric efficiency of

a motor provided with four valves in each cylinder instead of but two. This principle was thoroughly tried out in racing automobile motors, and is especially valuable in permitting of greater speed and power output from four- and six-cylinder engines, where cylinder bores are fairly large. On eight- and twelve-cylinder types, some engineers are doubtful if the resulting complication due to using a very large number of valves is worthwhile despite the wonderful results obtained when four valves are used especially in flat-head cylinders, where two valves of large size cannot very well be

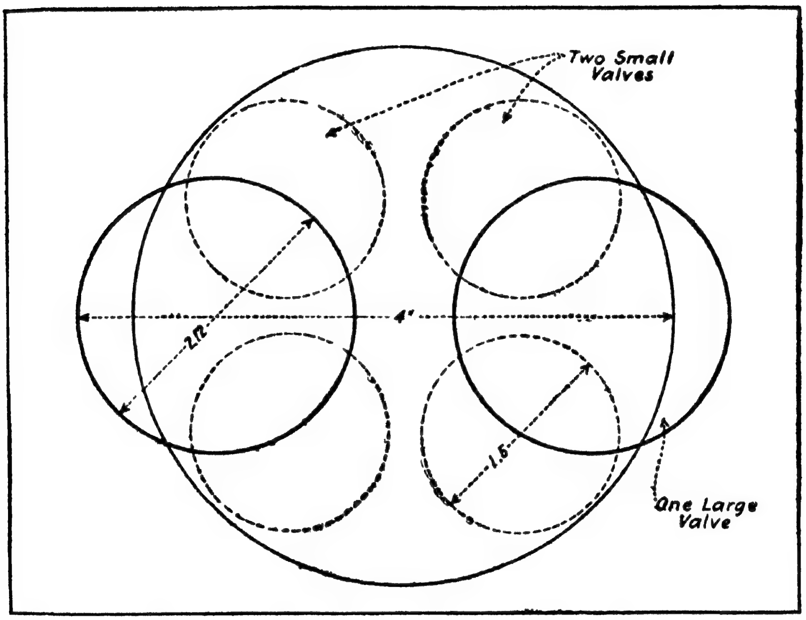


Fig. 278.—Diagram Comparing Two Large Valves and Four Small Ones of Practically the Same Area. Note How Easily Small Valves are Installed to Open Directly Into the Cylinder.

used. When extremely large valves are used, as shown in diagram at Fig. 278, it is difficult to have them open directly into the cylinder, and pockets are sometimes necessary. A large valve would weigh more than two smaller valves having an area slightly larger in the aggregate; it would require a stiffer valve-spring on account of its greater weight. A certain amount of metal in the valve-head is necessary to prevent warping; therefore, the inertia forces will be greater in the large valve than in the two smaller valves. As a greater port area is obtained by the use of two valves, the gases will be drawn into the cylinder or expelled faster than with a lesser area. Even if the areas are practically the same as in the diagram at Fig. 278, the smaller valves may have a greater lift without imposing greater stresses on the valve-operating mechanism and quicker gas intake and exhaust obtained. The smaller valves are not affected by heat as much as larger ones are. The quicker gas movements made possible, as well as reduction of inertia forces, permits of higher rotative speed, and, conse-

quently, greater power output for a given piston displacement.

Valve-Gears.—Most designs of spherical-head air-cooled cylinders necessitate valve-gears of rather freakish appearance. Two such examples, involving odd compound motions, are seen in the B.S.A. and Type J cylinders. It can only be said that such freakish gears have better mechanical properties than appearance and really function with considerable reliability. The motion in three planes, existing at the push-rod ball-end where it works in the rocker-cup, does not produce much trouble in practice. The use of an exposed push-rod valve-gear, in which lubrication is a matter of chance, is really crude in the extreme. Rocker-pivot lubrication can be made reason-

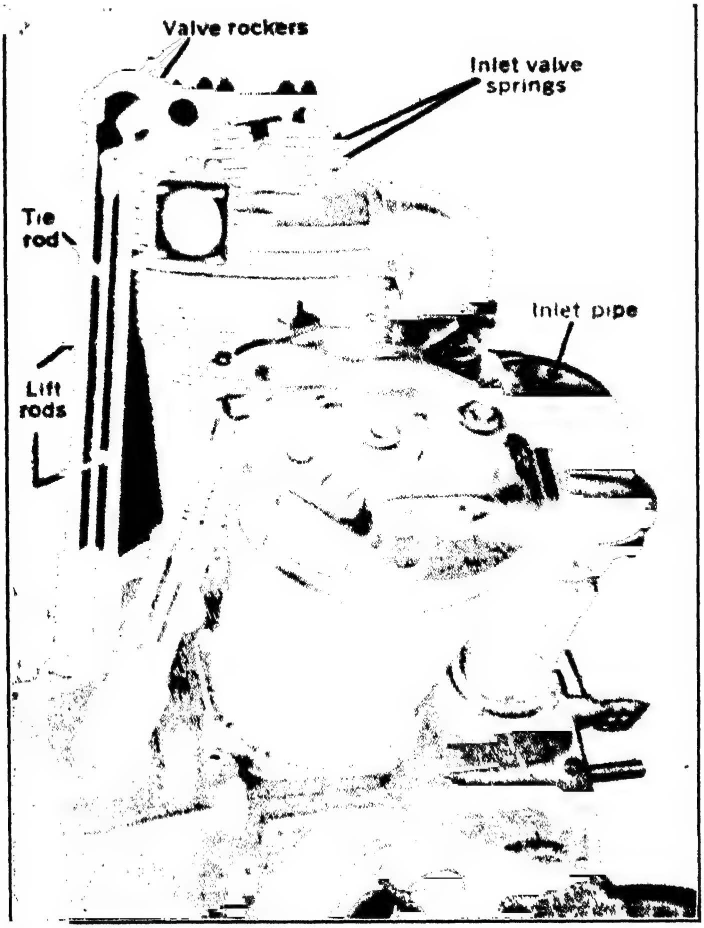


Fig. 279.—Compensation Arrangement Used on Bristol Jupiter Engines to Maintain Valve Clearances Uniform and to Allow for Cylinder Lengthening Due to Expansion When Heated.

ably satisfactory by special bearings and greases. The rocker ball-ends can be enclosed in gaiters. Provision can be made to maintain a constant tappet clearance by heating the push rods or by mechanical clearance compensation. For example, in the Bristol Jupiter engine a compensating ar-

rangement as shown at Fig. 279 is used. The two row large diameter four-lobed cam runs concentric with the crankshaft front end and is driven from it by eccentric epicyclic gearing at one-eighth engine speed in an anti-crank direction, operating by tappets and push rods the overhead rocker gear. The rocker gear is a special feature of the engine. The rockers are mounted on a bracket, which is secured at one end to the cylinder head and at the other end by a tie rod to the crankcase. This arrangement compensates for the radial expansion of the cylinders when hot and automatically maintains the desired valve clearances under all running conditions. However, when all these provisions are made, the result compares poorly with that of a fully enclosed valve-gear, in which the valve-stems, the springs, the rockers and the push rods are entirely enclosed and run in oil. A fully enclosed valve-gear avoids most of the objectionable shock, wear, noise and excessive change of tappet clearance and valve-timing encountered in the average open push-rod valve-gear. The air-cooled engine cannot hope to compete with the high-class water-cooled engine, if it be equipped with a noisy and rapidly wearing valve-gear, requiring almost daily adjustment and lubrication. In American radial-cylinder engines, the valve push rods are enclosed in tubes which act the same as the tension rod in the Jupiter engine. The valve-gear is thoroughly enclosed in modern engines and the compensation for cylinder lengthening due to heat is obtained as well. The cut away section of the Wright Whirlwind engine at Fig. 280 shows a modern American valve-gear of tried and proven design.

Rocker Compensating Gear.—On modern high efficiency air-cooled radial engines overhead valves are essential and usually operated by push rods through rockers mounted on the cylinder head. With conventional designs the resulting tendency is for the valve clearances to vary over a comparatively wide range as the cylinders heat up and expand, a typical variation of clearances being from 0.010 inch when cold to 0.060 inch when hot, that is under running conditions. In addition, even on similar engines the actual cylinder temperatures, and therefore the resulting clearances, will vary according to the installation and the operating conditions and climate. The result in practice is that the actual valve timing obtained is a variable quantity, and even in the most favorable cases usually a compromise setting, and the excessive clearances result in the valves, valve springs and rockers being subjected to excessively high accelerations and consequent increased wear, greater risk of failure and shorter life. All these drawbacks are overcome by the Patented Compensating Rocker Gear, used on "Bristol" Aero Engines, which automatically maintains the desired clearances within narrow limits, over the widest range of temperatures encountered in practice. This is one of the features contributing to the world-wide use and success of the Jupiter engine, and to its perfect functioning in Arctic cold and tropical heat, as borne out by actual independent official tests on standard engines. The rockers are mounted on the fulcrum pin carried in the rocker bracket, which is anchored to the cylinder head at its rear end and tied to the crankcase at its front end. As the engine warms up and the cylinder grows radially, the rocker bracket is held by the tie rod at the front end, about which it pivots, giving the variation of move-

ment between the rocker fulcrum and the cylinder required to obtain the desired compensating effect as shown at Fig. 279 A.

Jupiter—Tappet Assembly.—This unit affords an obvious illustration of the attention to detail characteristic of the Jupiter design. For the tappet guides, instead of the conventional phosphor bronze, duralumin is used with a substantial saving in weight. The thrust of the cams is taken by the hardened rollers running on hardened pins in hardened bearings completely immersed in oil spray, with the result that appreciable wear on these

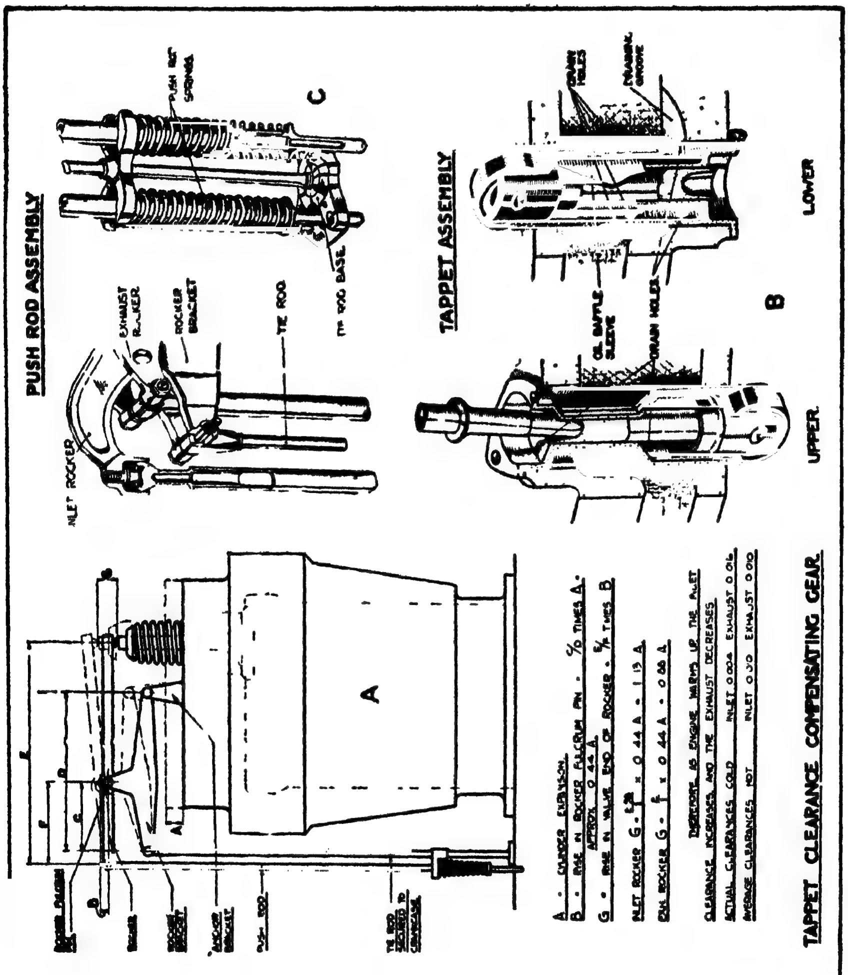


Fig. 279A.—Diagrams Showing Bristol Tappet Clearance Compensating Gear, also Push Rod and Tappet Assembly at B and C.

parts is never experienced. The cupped end of the tappet is radiused and hardened to receive the push rod end; allowing it perfect freedom to take up the correct alignment. On the tappets for the lower four cylinders the possibility of oil leakage is prevented by the special oil baffles which automatically trap and collect the surplus oil and drain it back into the crankcase. These components are clearly shown at B Fig. 279 A.

Jupiter—Push Rod Assembly.—The push rod tubes are fitted with special end pieces, hardened to prevent wear with it consequent alteration of valve clearances, and shaped to allow the rods perfect freedom to take up their correct alignment. The inner ends of both inlet and exhaust push rods are similar, but the outer ends differ, due to the fact that there are two exhaust rockers and one inlet rocker. The springs fitted to the inner ends of the push rods serve as auxiliary valve springs taking care of the inertia forces on the push rods and relieving the valve springs of all extra work. The construction is shown at C Fig. 279 A.

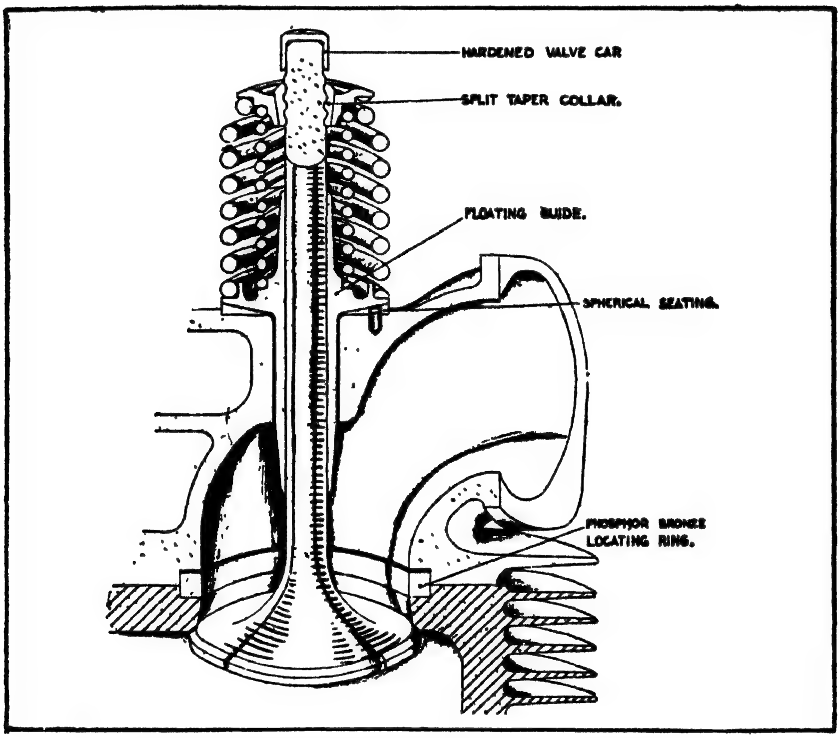


Fig. 279D.—Bristol-Jupiter Engine Valve Assembly.

Jupiter—Valve Assembly.—The spherical seating floating guides used for the inlet valves are a feature of British aero engine practice exclusive to the "Bristol" engines. Their use allows the valve to seat itself freely in spite of any distortion likely to occur under running conditions. From the maintenance point of view, an additional advantage is the ease with which replacements can be made, the guide being simply dropped in position. The twin valve springs, the maximum stress in each of which is only 60 per cent of the safe permissible stress, afford, in practice, absolute security against the risk of a fractured spring, allowing the valve to drop into the cylinder. The separate hardened valve caps taking the thrust of the rockers permit the most suitable valve material to be chosen unhampered by considerations of wear, and by reducing wear, ensure the maintenance of the desired valve clearances. The reader's attention is

directed to the sectional view at Fig. 279 D which shows how this design is worked out.

Few aircraft valve-gears entirely eliminate side-thrust on the valve-stem. Thus, when the relatively difficult bearing conditions of an exhaust valve-stem are considered, it is obvious why a hard valve-stem and guide are

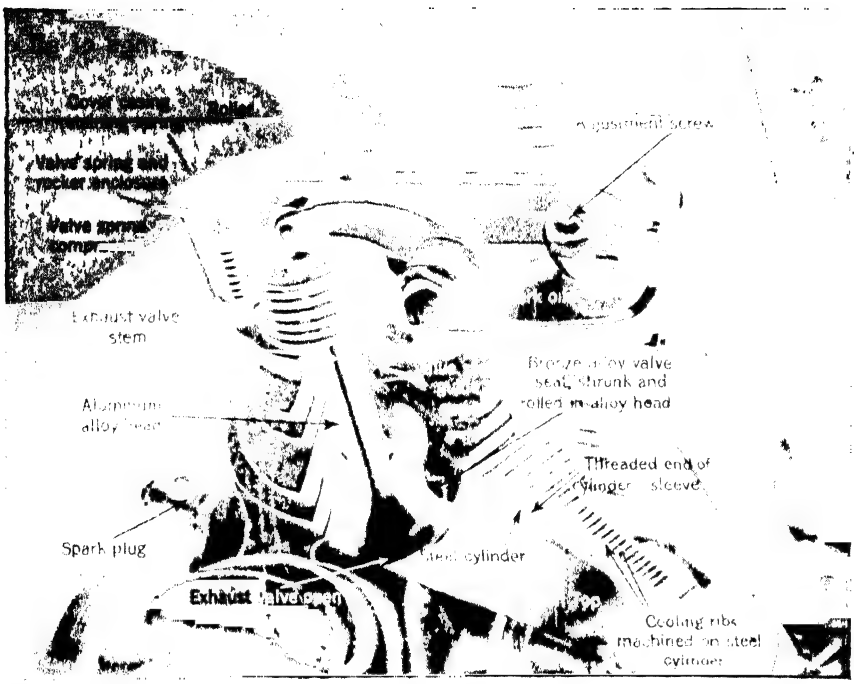


Fig. 280.—Valve Mechanism and Rocker Assembly of Wright "Whirlwind" Aviation Engine.

desirable. Even in cases where side-thrust on the valve-stems is entirely eliminated, scoring of the guide and the stem, particularly at the head end, occurs with soft valves and cast-iron guides. Rotation of the valve distributes the wear equally around the circumference of the stem and reduces or eliminates pitting of the valve-seat in the cylinder. Rotation can be produced by the volute ribbon type of valve-spring used on the Engineering Division air-cooled cylinders.

There is little question that a roller instead of a solid tappet on the end of the valve-rocker, where it bears on the valve-tip, has considerable effect in reducing side-thrust, thus reducing the wear of the valve-stem and the guide. That the roller as used in Wright engines and shown at Fig. 280 is of value is shown by its rapid rotation, apart from the observed reduction in wear. The larger the roller, within limits, the greater is its effect. Lubrication is of similar value. Neither of these devices, however, suffices to prevent scoring and wear with soft steel valves and cast-iron guides. The Type-K cylinder at first had soft stainless-steel valves and hard cast-iron guides. Despite a roller on the rocker end and ample lubrication,

valve-stem scoring and guide wear of both the inlet and the exhaust valves could not be obviated. An internally cooled valve of stainless steel that had been quenched and only drawn at 750 degrees Fahrenheit by the operation of filling with salt gave much better results than the soft stainless-steel valves, showing only very slight scoring, but was inferior to a hard tungsten-steel valve.

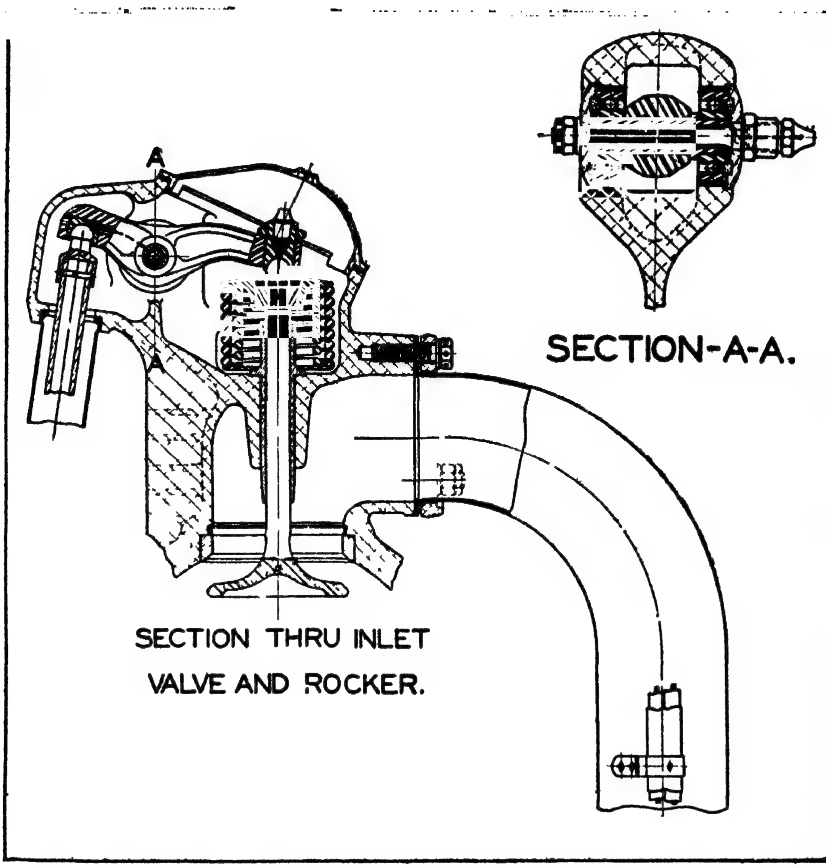


Fig. 281.—Section Through Inlet Valve Rocker of Pratt & Whitney "Wasp" Engine.

Valve Gear Enclosure.—The very complete valve enclosure provided on the Pratt and Whitney Wasp engines is also worthy of comment. A sectional view showing the inlet valve and rocker is given at Fig. 281 and a similar view outlining the construction of the exhaust valve and rocker is presented at Fig. 282. It will be observed that lubrication of the ball-end valve-stem as well as the ball end of the push rod is accomplished by Zerk oilers, one placed above the valve-stem, the other on the rocker arm bearing bolt. The rocker bearings are of the ball type and will function with minimum oiling. The valve-spring housings are cast integrally with the cylinder-head as shown at Fig. 283 instead of being separate boxes as in the J5

Whirlwind engines. The valve mechanism is enclosed with pressed metal covers retained by a self-locking spring bail arrangement which permits of ready access by removing the cover and yet thorough enclosure when the cover is in place.

Packard Oil-Cooled Valves.—One of the important features in the design of large aircraft engines is the manner of valve cooling, especially the exhaust valves. In the Packard engines this is accomplished by circulating oil through the valve-stem as shown at Fig. 284. Means for cooling the exhaust valves by the circulation of oil are provided by suitably drilled

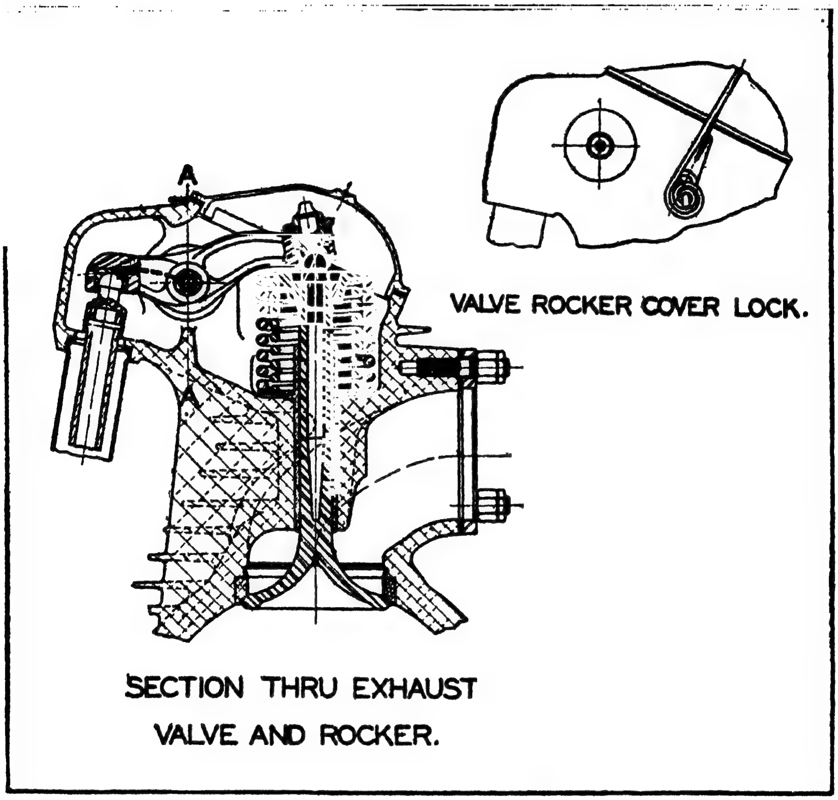


Fig. 282.—Section Through Exhaust Valve Rocker of Pratt & Whitney "Wasp" Engine.

passages in the camshaft bearings adjacent to the exhaust cams, this feature of the construction being shown diagrammatically. The camshaft is hollow and is supplied with oil under pressure through a continuous metering groove in the rear bearing. In the camshaft journal next to each exhaust cam is drilled a hole opposite to the nose of the corresponding exhaust cam. This hole registers with a vertical passage in the.camshaft bearing pedestal when the cam is at its highest point and the exhaust valves are closed. The oil flows through this passage to the bottom of the cam follower guide, which forms a closed cylinder, the space under the cam follower being thus filled with oil. The camshaft in revolving cuts off communication with this

passage and when the cam follower is depressed by the cam, the oil can escape only by being forced through the hollow cam follower stem and the horizontally drilled passages leading out through the drilled tappets into the exhaust valve-stems. The latter are drilled throughout their entire length, the lower end of the hole in the valve head being closed by a screwed-in plug. A small steel tube is welded to this plug and is centered in a counter-bore at the upper end of the valve-stem. The oil is forced down through

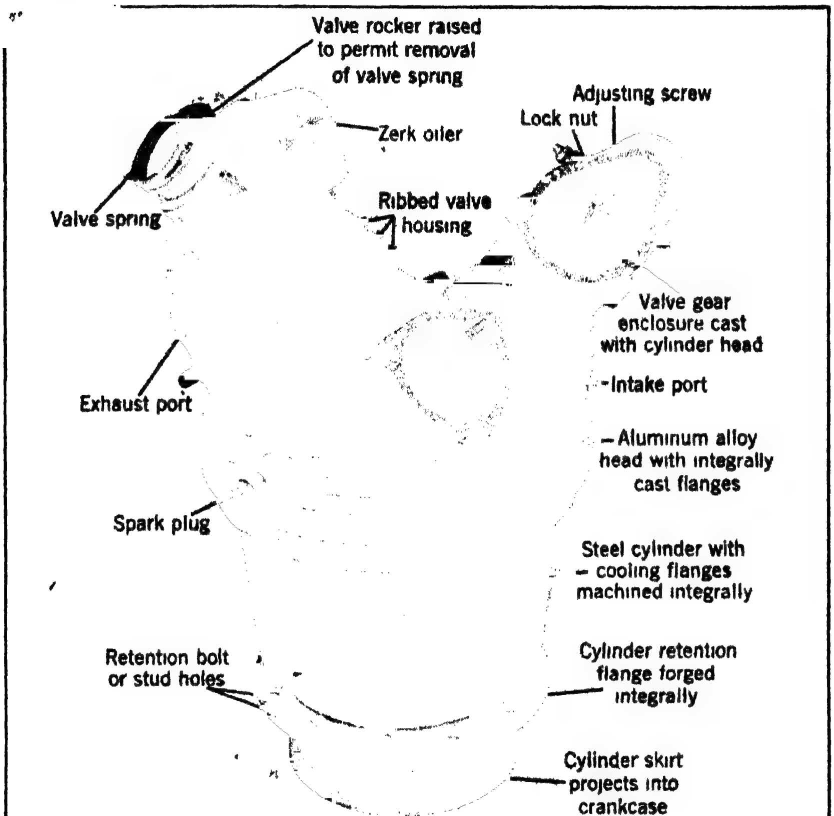


Fig. 283.—Pratt & Whitney "Wasp" Engine Cylinder Construction Showing Integrally Cast Rocker Arm Enclosing Housing. Note Large Intake and Exhaust Ports and Method of Removing Valve Spring.

the tube and out of the bottom through horizontal holes, thus cooling the head of the valve. The oil is discharged through the annular space between the tube and the inner wall of the valve-stem and out of the valve housing through horizontal holes drilled through the upper end of the valve-stem just below the counterbore. As a result of this oil-cooling the exhaust valves operate at very low temperatures and the valve seat is maintained in good condition for long periods. The problem of valve cooling is of special importance when considered in connection with air-cooled cylinder design as the oil-cooling method just outlined would not be practical with air-cooling. This matter will be considered in more detail as it relates to

air-cooled engines when we consider the design of this form of cylinder which introduces problems calling for an entirely different solution than possible with water-cooled engines.

Bore and Stroke Ratio.—A question that has been a vexed one and which has been the subject of considerable controversy is the proper proportion of the bore to the stroke. The early gas-engines had a certain well-defined bore to stroke ratio, as it was usual at that time to make the stroke twice as long as the bore was wide, but this cannot be done when high speed is desired. With the development of the present-day motor the stroke or

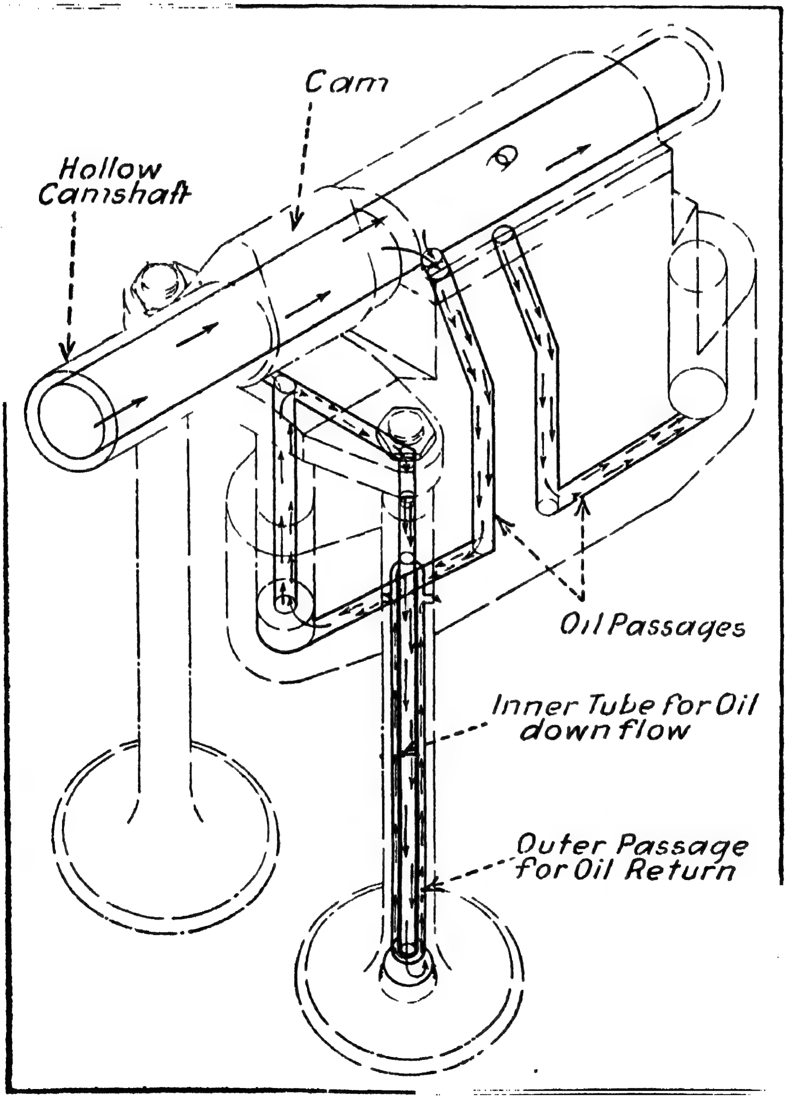


Fig. 284.—Diagram Showing Oil Circulation Through Exhaust Valves of Packard Aircraft Engines Models 2A-1500 and 2A-2500 to Keep Valves Cool.

piston travel has been gradually shortened so that the relative proportions of bore and stroke have become nearly equal. Of late there seems to be a tendency among designers to return to the proportions which formerly obtained, and the stroke is sometimes one and a half or one and three-quarter times the bore.

Engines designed for high speed should have the stroke not much longer than the diameter of the bore. The disadvantage of short-stroke engines is that they will not pull well at low speeds, though they run with great regularity and smoothness at high velocity. The long-stroke engine is much superior for slow speed work, and it will pull steadily and with increasing power at low speed. It was formerly thought that such engines should never turn more than a moderate number of revolutions, in order not to exceed the safe piston speed of 1,000 feet per minute. This old theory or rule of practice has been discarded in designing high efficiency automobile racing and aviation engines, and piston speeds from 2,500 to 3,500 feet per minute are sometimes used, though the average is around 2,000 feet per minute. While both short- and long-stroke motors have their advantages, it would seem desirable to average between the two. That is why a proportion of four to five or six seems to be more general than that of four to seven or eight, which would be a long-stroke ratio. Careful analysis of a number of foreign aviation motors shows that the average stroke is about 1.2 times the bore dimensions, though some instances were noted where it was as high as 1.7 times the bore.

One of the problems that confront the automotive engineer every time a new engine is designed is that of the best proportion of stroke to bore.

What is meant by "best" in this connection calls for some explanation. Whenever a change is made in the stroke-bore ratio several characteristics of the engine are affected. Included among these is the weight efficiency, that is, the weight of the complete engine per unit of maximum power output. Other factors that are subject to change with the stroke-bore ratio are the thermal efficiency (fuel economy) and the smoothness of operation or freedom from vibration. Under certain conditions the reliability of the engine in service and its useful life may also be affected, but it is evidently the most logical plan to so proportion the parts that the stresses in the various members and the unit pressures on all bearing surfaces remain the same, in which case there is no reason to expect any change in the life or in the degree of reliability. Evidently a weighing of the effects of a change in the stroke-bore ratio on all of these factors is necessary in order to determine what is the best ratio from what may be called the operating efficiency standpoint.

Mr. P. M. Heldt, M.S.A.E., writing in *Automotive Industries* has shown that if two engines are built generally along the same lines, both having the same piston displacement but one a stroke equal to twice its bore while in the other the stroke and bore are alike—

(1) Both engines will develop the same power at the same speed.

(2) The long-stroke engine will weigh slightly more than the short-stroke one.

(3) At the same speed the inertia forces are slightly greater in the long-stroke engine.

(4) At the same speed, unit loads on piston pin and crankpin bearings due to inertia are materially smaller in the short-stroke engine.

(5) With crankshafts of about the same diameter, corresponding critical speeds are higher in the short-stroke engine.

(6) It is possible to use much larger valves in the short-stroke engine.

(7) The short-stroke engine can be run safely at higher speeds of revolution and develop more power.

(8) From the standpoint of fuel economy the long stroke may have a slight advantage, but hardly sufficient to be detected by means of commercial instruments.

Meaning of Piston Speed.—The factor which limits the stroke and makes the speed of rotation so dependent upon the travel of the piston is piston speed. Lubrication and inertia forces are the main factors which determine piston speed, and the higher the rate of piston travel the greater care must be taken to insure proper oiling and the lighter the reciprocating parts must be. Let us fully consider what is meant by piston speed. Assume that a motor has a piston travel or stroke of six inches, for the sake of illustration. It would take two strokes of the piston to cover one foot, or twelve inches, and as there are two strokes to a revolution it will be seen that this permits of a normal speed of 1,000 revolutions per minute for an engine with a six-inch stroke, if one does not exceed 1,000 feet per minute the figure formerly considered desirable but now greatly exceeded and without serious or harmful results though with great increase of power output for a given piston displacement. If the stroke was only four inches, a normal speed of 1,500 revolutions per minute would be possible without exceeding the prescribed limit. The crankshaft of a small engine, having three-inch stroke, could turn at a speed of 2,000 revolutions per minute without danger of exceeding the safe speed limit. It will be seen that the longer the stroke the slower the speed of the engine, if one desires to keep the piston speed within the bounds as recommended, but modern practice allows of greatly exceeding the speeds formerly thought best by mechanical engineers.

Present-day automotive engines have been divided into four classes with respect to piston speed by Mr. P. M. Heldt. The slowest are those used for heavy trucks and tractors, which are also used to a certain extent for industrial work. Next comes the four-cylinder passenger car engine. This is followed by the six- and eight-cylinder passenger car engine, and finally comes the modern supercharged racing engine. Average piston speeds of the four classes are about as follows:

Truck and tractor engines	1300 ft. p.m.
Four cylinder passenger car engines	1600 ft. p.m.
Six and eight cylinder passenger car engines	2200 ft. p.m.
Racing engines	3400 ft. p.m.

The spreads between the different classes appear larger when the speeds are given in revolutions per minute, for the reason that the low-speed engines have longer strokes than the high-speed engines. A comparison of the four classes on the r.p.m. basis is given in the following table.

Four cylinder truck and tractor engines	1050 r.p.m.
Four cylinder passenger car engines	2250 r.p.m.
Six and eight cylinder passenger car engines	2800 r.p.m.
Racing engines	6800 r.p.m.

Aviation Engine Crankshaft Speeds.—It will be evident that most aviation engines designed to drive a direct drive propeller turn slower than four-cylinder passenger car engines and very few geared engines, except very small ones, exceed the rotative speed of six- and eight-cylinder passenger car engines. Anzani engines range in speed from 1,600 to 1,800 r.p.m. The Curtiss powerplants turn as follows, OX5 at 1,400 r.p.m.; OXX6, 1,900 r.p.m.; C6A, 1,700 r.p.m.; D12, 2,300 r.p.m.; the V1570 at 2,400 r.p.m. The B2 Salmson Menasco turns its crankshaft at from 1,500 to 1,700 r.p.m. The Packard engines turn as follows at sea level: the 1A2775 X type at 2,700 r.p.m.; the 3A1500 at 2,700 r.p m. maximum, the 3A2500 at 2,100 r.p.m. Both Pratt and Whitney Wasp and Hornet motors turn at 1,900 r.p.m. The Wright Whirlwind J5 has a maximum rotative speed of 2,000 r.p.m., the Cyclone R1750 turns at 1,900 r.p.m. Some small engines, such as the Bristol Cherub, have rotative speeds of 3,000 r.p.m. but no aviation engine has yet approached supercharged automobile racing engines in crankshaft speed as a recently developed motor of this character has reached 8,000 r.p.m.

The chief object of increasing the piston speed or the speed of revolution is, of course, to obtain greater power from an engine of given displacement, and, therefore, from a given engine weight. How the output per unit of displacement is increased by increasing the piston speed is clearly shown by the following tabulation prepared by Mr. Heldt which gives the reciprocal values, the displacements required to produce one horsepower. The figures given are averages for practically all engines of each class now on the American market. To produce one horsepower requires—

11 cubic inches in truck, tractor and industrial engines of over 500 cubic inch displacement
6 cubic inches in four-cylinder passenger car engines
3.75 cubic inches in six- and eight-cylinder passenger car engines
0 59 cubic inches in supercharged racing engines.

Analyzing some published figures of horsepower output and piston displacement we find that all aviation engines are superior to automobile engines. That having the greatest displacement per horsepower, i.e., the Curtiss OX5 is the one more nearly approximating automobile practice. The OX series give about one horsepower for each five inches displacement. A great improvement is noted in later Curtiss designs however, as we find one horsepower for each 2.8 cubic inches in the D12 and one horsepower for each 2.5 cubic inches in the V1570 engine. The Packard engines range from 2.3 cubic inches per horsepower in the 1A2775 to one horsepower per three cubic inches in the other types. The Pratt and Whitney Wasp shows one horsepower for each 3.3 cubic inches as does the Hornet engine of the same manufacture. The Wright J5 delivers one horsepower for each 3.5 cubic inches of piston displacement. The difference between the Wright J5 and the Pratt and Whitney engines may be ascribed to the use of a supercharger by the latter.

The high efficiency value obtained by racing auto engines of 0.59 cubic inch per horsepower cannot be reached by aviation engines for two reasons, the most important being that the high r.p.m. is not practical in the light of our present knowledge owing to gear reduction drive difficulties with avia-

tion engines and also due to the fact that the supercharger of an aviation engine is used primarily to maintain engine power at high altitudes where the air is thin and not for increasing power at sea level because airplanes fly at high altitudes.

Mr. Heldt gives the following average brake mean effective pressures for engines of the different classes:

68.5 pounds per square inch for truck, tractor and industrial engines of over 500 cubic inch displacement
59 pounds per square inch for four-cylinder passenger car engines
76 pounds per square inch for six- and eight-cylinder passenger car engines
197 pounds per square inch for supercharged racing engines.

The range of brake mean effective pressures is higher in aviation engines than in automobile engines because higher compression is employed and values as high as 135 pounds per square inch are realized in practice though the average value will be nearer 120 pounds per square inch than the higher figure. This value is based on compression pressure rather than rotative or piston speeds exclusively, as compression before ignition is an important factor in securing high brake mean effective pressure.

Crankshaft Vibration Limits Speed.—To prevent crankshaft vibration due to increased r.p.m. we must increase the shaft diameter and thus raise the natural speed of torsional vibration of the shaft. The critical speed varies substantially as the square of the crankshaft diameter, and, according to Mr. Heldt, if the speed range of the engine is increased, and it is desired that the relation of the different critical speeds to the speed range remain the same, the crankshaft diameter must be increased in the proportion of the square root of the speed increase. Actually the diameter should increase somewhat faster than the square root of the speed, for the reason that when the main journals are increased in diameter the crankpins and crankarms are also increased in size and this adds to the vibrating mass. The magnitude of this influence has to be estimated, but it is probably not far wrong to assume that in order to keep the relation between different critical speeds and the speed range the same, the crankshaft main bearing diameter must vary about as the 0.6th power (instead of the 0.5th) of the speed. Therefore, if the speed of the engine is doubled the crankshaft diameter must be increased in the proportion of (roughly) 1.5 to 1.

Inertia Forces Increase with Speed.—One of the most serious results of an increase in engine speed is the accompanying increase in inertia and centrifugal forces. If the reciprocating and rotating weights remain the same, then these forces increase as the square of the speed, and, therefore, much faster than the power. These forces subject the members on which they act, and adjacent members, to additional stresses, and by increasing the loads on the bearings, they add to the frictional losses. Considering for the moment only the inertia forces, they depend chiefly on the speed of rotation, increasing as the square of this factor, and also on the length of stroke and the mass of the reciprocating parts. If the speed is to be increased, the only way in which the inertia force can be kept down is by reducing the mass of the reciprocating parts.

In the higher speed engines the reciprocating parts therefore must of necessity be lightened. As regards the pistons, this can be done either by cutting down the thickness of the skirt to a minimum and reducing the length, or by casting the piston of light alloy. As regards the connecting rod, its weight can be lessened by making it of alloy instead of carbon steel and by machining it all over. In some engines of recent development forged duralumin connecting rods have been successfully employed. All of these changes from conventional practice for moderate speed engines involve additional cost. There are, of course, limits to the reduction in weight of both piston and connecting rods, and with very high-speed engines the inertia forces and consequent bearing loads will inevitably be higher.

Bearings Heat at High Speed.—At very high speeds it is difficult to prevent the bearings—especially the crankpin bearings—from overheating. The generation of heat in plain bearings depends upon the rubbing speed and also upon the load on the bearing. When the engine speed is increased, that in itself increases the rubbing speed. The latter is further increased by the fact that if the engine is to be run at materially higher speed its crankshaft and crankpin bearings must be made larger in diameter. Finally, the load on the bearing is increased by the increase in the inertia forces. Ball and roller main bearings can be operated at higher speeds and under heavier loads without heating or friction losses than plain bearings, though their size and weight compared to a plain babbitt bearing precludes their use in connecting-rod big-ends of very high-speed engines. The speed of modern aviation engines today is limited by: (*a*) breathing capacity, (*b*) the dissipation of heat from the connecting-rod big-end bearings, and (*c*) the mechanical operation of the valves. The breathing capacity can be almost doubled, if necessary, by employing a double row of valves along the engine and by using two camshafts, but so doing brings the effect of (*b*) and (*c*) into play. By using roller-bearings or floating bushings, the friction and heat-flow from the big-end bearings can be minimized; and, by mounting the camshafts directly against the valves, the mass to be accelerated can be reduced. By such means the maximum power can, if necessary, be attained at piston speeds exceeding 4,000 feet per minute; but the piston speed of the simplest form of high-speed engine—with plain bearings and a single camshaft as near the crankshaft as possible—is limited to about 2,500 feet per minute, for maximum power.

The crankpin bearing load also depends to a considerable extent upon the centrifugal force on the connecting-rod head, which also increases as the square of the speed of rotation. This, to a certain extent, is an important factor limiting the speed of radial or W engines where three or more connecting rods terminate on one crankpin bearing. In this case, too, the only means of keeping down the force is by lightening the member, and a considerable improvement has been made in this respect by spinning the babbitt lining directly into the head instead of fitting removable bearing shells. This is probably the only instance where lightening of the moving parts has not resulted in increased cost. The spun-in bearing has the further advantage that there is no break in the path of heat flow from the bearing surface to the metal of the rod.

Off-Set Cylinders.—Another point upon which considerable difference of opinion existed related to the method of placing the cylinder upon the crankcase—i.e., whether its center line should be placed directly over the center of the crankshaft, or to one side of center. The motor shown at Fig. 285 is an off-set type, in that the center line of the cylinder is a little

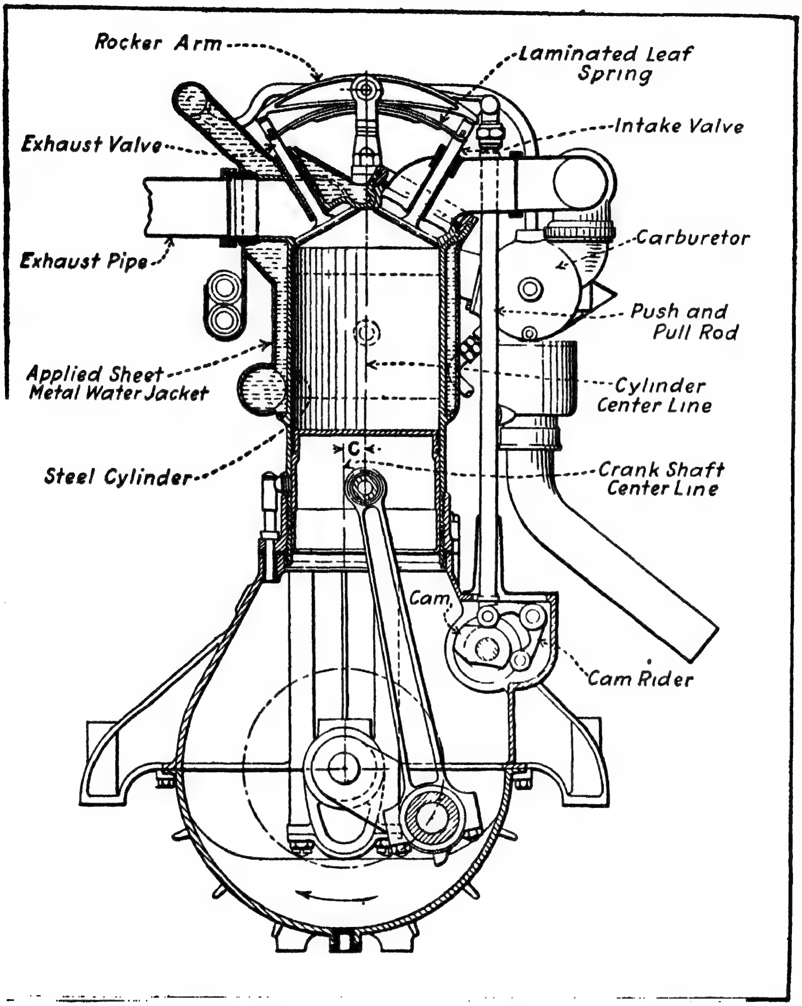


Fig. 285.—Cross Section of Early Austro-Daimler Engine Showing Off-Set Cylinder Construction. Note Peculiar Valve Action and Use of Laminated Leaf Spring to Return Valves to their Seating.

to one side of the center of the crankshaft. Diagrams are presented at Fig. 286 which show the advantages of off-set crankshaft construction. The view at A is a section through a simple motor with the conventional cylinder placing, the center line of both crankshaft and cylinder coinciding. The view at B shows the cylinder placed to one side of center so that its center line is distinct from that of the crankshaft and at some distance from

it. The amount of off-set allowed is a point of contention, the usual amount being from fifteen to 25 per cent of the stroke. The advantages of the off-set are shown at Fig. 286 C. If the crank turns in direction of the arrow there is a certain resistance to motion which is proportional to the amount of energy exerted by the engine and the resistance offered by the load. There are two thrusts acting against the cylinder wall to be considered, that due to explosion or expansion of the gas, and that which

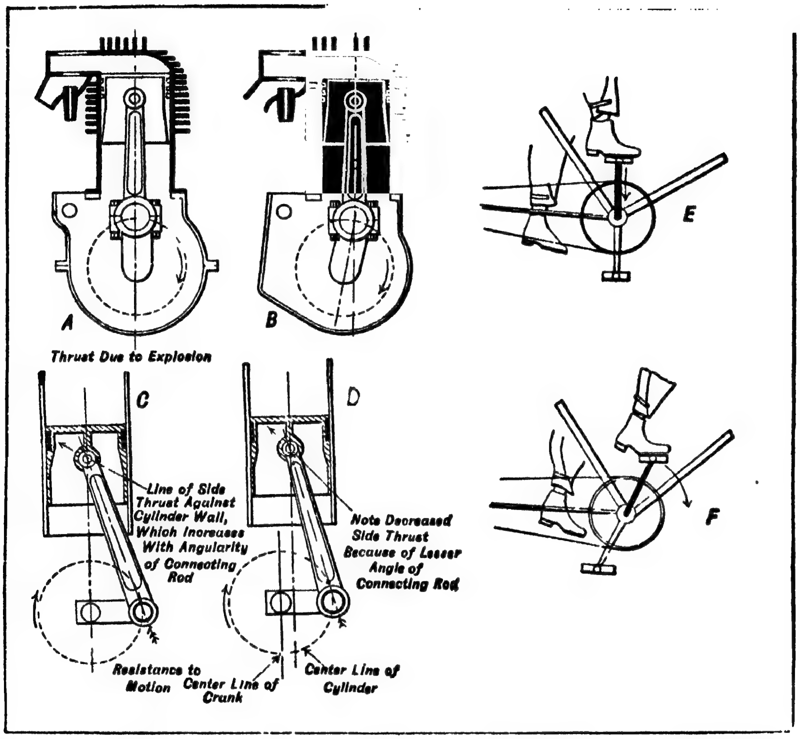


Fig. 286.—Diagrams Demonstrating Advantages of Off-Set Crankshaft Construction.

resists the motion of the piston. These thrusts may be represented by arrows, one which acts directly in a vertical direction on the piston top, the other along a straight line through the center of the connecting rod. Between these two thrusts one can draw a line representing a resultant force which serves to bring the piston in forcible contact with one side of the cylinder wall, this being known as side thrust. As shown at C, the crankshaft is at 90 degrees, or about one-half stroke, and the connecting rod is at twenty degrees angle. The shorter connecting rod would increase the diagonal resultant and side thrusts, while a longer one would reduce the angle of the connecting rod and the side thrust of the piston would be less. With the off-set construction, as shown at D, it will be noticed that with the same connecting-rod length as shown at C and with the crankshaft at 90 degrees of the circle that the connecting-rod angle is fourteen degrees and the side thrust is reduced proportionately.

Another advantage claimed is that greater efficiency is obtained from the explosion with an off-set crankshaft, because the crank is already inclined when the piston is at top center, and all the energy imparted to the piston by the burning mixture can be exerted directly into producing a useful turning effort. When a cylinder is placed directly on a line with the crankshaft, as shown at A, it will be evident that some of the force produced by the expansion of the gas will be exerted in a direct line and until the crank moves the crank throw and connecting rod are practically a solid member. The pressure which might be employed in obtaining useful turning effort is wasted by causing a direct momentary pressure upon the lower half of the main bearing and the upper half of the crankpin bushing. Very easily understood illustrations showing advantages of the off-set construction are shown at E and F. This is a bicycle crank-hanger. It is advanced that the effort of the rider is not as well applied when the crank is at position E as when it is at position F. Position E corresponds to the position of the parts when the cylinder is placed directly over the crankshaft center. Position F may be compared to the condition which is present when the off-set cylinder construction is used.

QUESTIONS FOR REVIEW

1. Outline common methods of airplane motor cylinder construction.
2. What are advantages of block construction?
3. Why is combustion-chamber form important?
4. What is the difference between wet and dry sleeve cylinder construction?
5. Name common methods of locating valves in automotive engine cylinders.
6. Why is the I head form used on most airplane engines?
7. How are valves in L head cylinders operated?
8. Describe various forms of overhead valve gear.
9. Why are multiple valve springs advantageous?
10. How do springless valves work?
11. Describe Knight sleeve valve construction.
12. How does a single sleeve valve work?
13. What is the best construction for aero engine valves?
14. What is the effect of gas velocity on power output?
15. Describe a typical rocker compensating gear and give reasons for its use.

CHAPTER XX

AIR-COOLED CYLINDER CONSTRUCTION

Air-Cooled Cylinder Design—Temperature Distribution in Air-Cooled Cylinder—Effects of Cooling Air Supply—Quantity of Air Needed for Cooling—Effect of Mixture Strength on Cooling—Effect of Air Blast Direction—Air-Cooled Cylinder Forms—Cooling Fin Dimensions—Law of Heat Radiation—Heat Flow in Fins—Rectangular Fins—Rates of Heat Dissipation—Circumferential Finning Best—Types of Air-Cooled Cylinder Heads—How Auto and Aviation Practice Differ—Large Cylinders Air-Cooled—Spherical and Roof Heads—Composite Cylinder Construction—Alloy Head Cast on Steel Barrel—Bolted-on Separable Heads—Cast Cylinder of Alloy with Liner—Steel Barrel with Alloy Cap—Materials for Air-Cooled Cylinders—Alloys for Cylinder Heads—Cast Iron for Cylinders—Improved Method of Melting Cylinder Iron—Nichrome Improves Cylinder Iron.

A great amount of research work has been done by S. D. Heron, M.S.A.E., who for a time was an aeronautical mechanical engineer at McCook Field, Dayton, Ohio, on air-cooled cylinders and this work has supplemented previous work carried on in England by Dr. Gibson of the Royal Aircraft Establishment, in which Mr. Heron assisted. This subject was covered at length and in considerable detail in a paper read before the Dayton section of the S.A.E. and published in the Journal of the Society in April, 1922.

Air-Cooled Cylinder Design.—Investigation by Mr. Heron has shown that for every brake horsepower developed an average of approximately 0.6 horsepower or 25 B.t.u. per minute has to be dissipated directly to the cooling air by the external cooling surface of the cylinder. In addition, 0.4 to 0.5 horsepower has to be dissipated by the oil, by conduction to and radiation from the crankcase and similar means. The amount of heat absorbed by the oil will depend largely upon the amount reaching the cylinder and the piston walls and the facilities for cooling the oil. Power output, fuel consumption and cylinder-wall temperatures, such as are quoted herein, are dependent upon liberal splash lubrication and the resultant oil-cooling.

In the light of present-day knowledge a design for a cylinder of high output has to fulfill approximately the following requirements:

(1) Develop a volumetric efficiency of 80 to 85 per cent
(2) Produce a brake mean effective pressure of at least 130 pounds per square inch at the normal speed on a maximum fuel-consumption of 0.56 pound per brake horsepower-hour
(3) Dissipate 25 B.t.u. per minute per brake horsepower from the external cooling surfaces of cylinder, this heat to be dissipated so that the maximum temperature of any portion of the exterior of the cylinder walls does not exceed 550 degrees Fahrenheit, and it is preferable that it be lower. In addition, the mean temperature of the exterior of the cylinder walls should not exceed 350 degrees Fahrenheit. To produce a layout that fulfills the stated requirements, it is necessary that heat-flow be the primary point in mind during design.

Temperature Distribution in Air-Cooled Cylinder.—As the amount of heat lost to the walls from the charge differs largely in various parts of the cylinder, it is obvious that to have anything like even temperature-distribution, the supply of cooling air to any portion of the cylinder should be proportioned approximately to the amount of heat given to that portion of the cylinder. Considering a normal design of overhead-valve cylinder with circumferential cooling-fins, it is evident that the side of the cylinder-head and barrel that carries the exhaust ports will receive the greatest heat supply per unit area, and therefore should receive the major portion and the greatest effect of the cooling air supply. This requirement is suitably met by applying the cooling blast on the exhaust side of the cylinder. In practice, with such blast application, the circumferential temperature-difference at the top of the cylindrical portion of the combustion-chamber will not exceed 50 degrees Fahrenheit. Toward the base of the cylinder the circumferential temperature-differences will probably increase, but this is usually of little moment, since the maximum temperatures attained there are low. Contrary to the opinion commonly held in this country and often advanced by opponents of air-cooled engines the back or side of the cylinder that is in the lee of the blast, does not give overheating trouble when the cylinder design is sound and the air supply is suitably arranged.

Uneven temperature distribution, whether caused by poor air distribution or by cylinder walls lacking the required heat-flow capacity to equalize the temperature distribution, has a considerable effect on the output, thermal efficiency and reliability of a cylinder. Very uneven distribution of temperature will result in the local development of an excessive temperature and overheated exhaust-valves. These faults in their turn lead to decreased volumetric efficiency and the use of rich mixtures to check detonation, to reduce the flame temperature and to cool the walls internally.

Effects of Cooling Air Supply.—The temperature of an air-cooled engine is determined by the cooling-air temperature as a datum point. If the air temperature rises 50 degrees Fahrenheit, the actual cylinder temperatures are sensibly increased by that amount. In general, with an efficient cylinder design, the effect of the air temperature is little felt. A considerable rise in the air temperature increases the cylinder temperature, but it simultaneously reduces the charge weight, and thus to some extent the heat to be dissipated by the cylinder is diminished. A variation of 350 degrees Fahrenheit in maximum cylinder-temperature at full throttle is permissible for short periods of time, such as exist during a fast steep climb by an airplane or the ascent of a mountain pass by an air-cooled car. That cylinder-temperature control is necessary for air-cooled aircraft engines has yet to be proved in practice, although urged as a disadvantage of air-cooling by its opponents. The stabilization of the carburetor temperature is much more likely to be found necessary, due to the rapidity with which an air-cooled engine cools during a glide or dive when switched off or idling compared to the cooling down of a water-cooled cylinder under similar conditions.

An investigation of the amount of air required to carry away the heat dissipated from the external cooling surfaces of air-cooled cylinders is of considerable interest. For aircraft it is desirable to use as little cooling air

as possible to minimize the head resistance. For automobile work, where fan or blower cooling is used, economy of power absorbed by the fan is to be aimed at. In both cases, however, economy in the cooling air supply may be dearly bought. An inevitable result of insufficient cooling air supply is a hot engine, requiring more fuel for a given net performance than an engine supplied with more cooling air, and having a lower specific fuel-consumption per overall horsepower which is the effective horsepower plus the horsepower absorbed in cooling. Experience, however, has shown that to produce an air-cooled cylinder that is comparable in output with a high-class water-cooled design, the cooling medium must flow over all

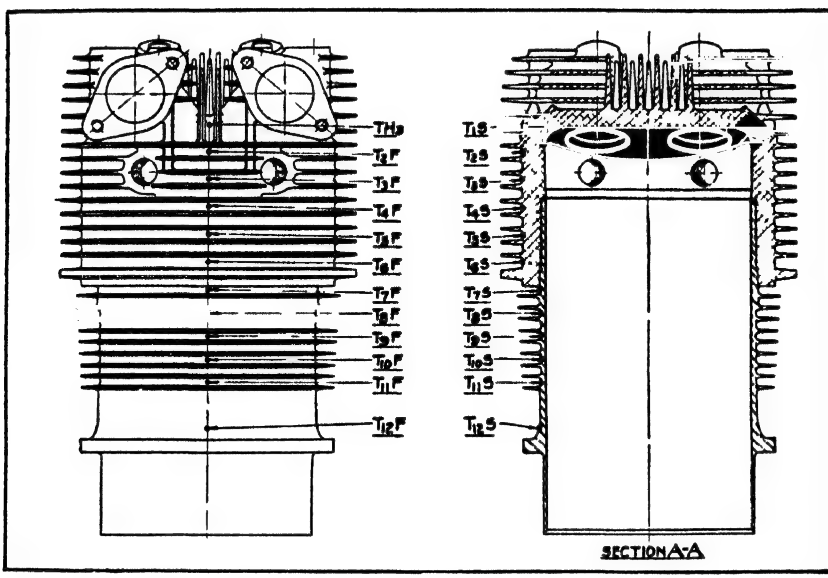


Fig. 287.—Temperature Readings at Various Positions of the Airco Aircraft Engine Cylinder.

portions of the cylinder-head and barrel. Air will not flow around sharp corners or through fins at 90 degrees to the airstream merely to fit in with the imagination of the sanguine designer. Numerous tests have demonstrated that siamesed exhaust-ports, and exhaust and intake ports without any air space between them, are altogether unsound. This is markedly comparable with experience on water-cooled cylinders of high output, but water is like the broad mantle of charity, it covers a multitude of sins. Defects in design that are not apparent in water-cooled engines are clearly indicated by reduced efficiency of an air-cooled cylinder.

The aim in design should be to remove as far as possible the heat from the cylinder at the point where it is given to the head, ports and barrel. Investigation has demonstrated that neither a material of high conductivity nor an excessive cooling air-supply will remedy poor design. A simple cast-iron cylinder with only a small cooling air-supply but fulfilling some of the fundamental necessities for efficient heat-dissipation will

give a performance much superior to that of designs that presume to function by high wall or fin conductivity in conjunction with large quantities of air supplied to those portions of the cylinder where little is needed and a total lack of effective air-supply where it is required.

An attempt was made by Mr. Heron to estimate the quantity of air used for cooling a circumferential-fin cylinder by a free, unconfined, blast. The method used is not exact by any means and is open to criticism, but

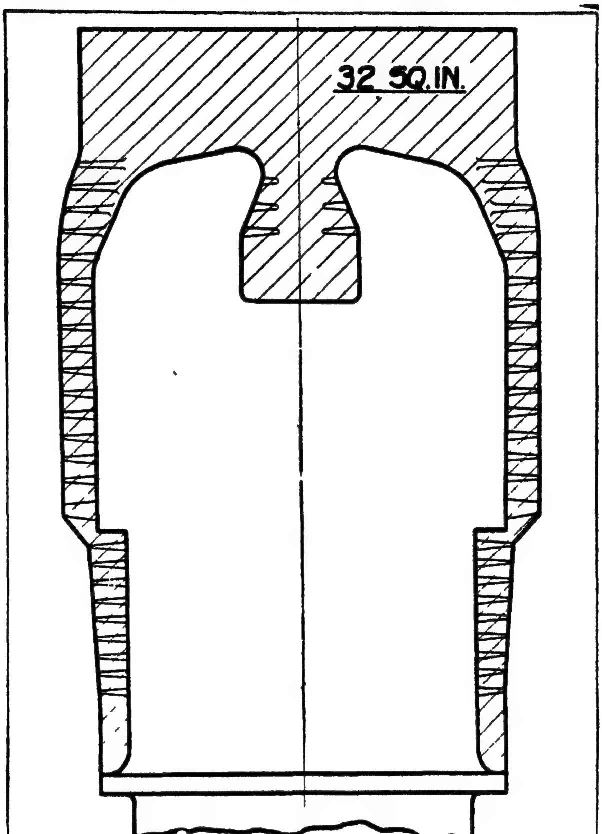


Fig. 288.—Cross Section Showing Available Area for Useful Cooling Air Flow Provided by the Airco Cylinder.

other data on cylinders of such size and efficiency are lacking. The cylinder used in the tests was the Airco of 5½ by 6 inches, shown in Fig. 287 developing 46.5 brake horsepower at 1,900 r.p.m., or a brake mean effective pressure of 136 pounds per square inch, and running in a mean blast of 87 m.p.h., this velocity being the mean of the velocities measured around the outside of the air-flow area contour shown in Fig. 288. The cylinder is assumed to be cooled so that all air supplied passes between the fins. It is assumed also that the air velocity is constant from the roots to the tips of the fins. The air-flow area diagram takes no account of the fin thickness and otherwise gives a generous estimate of area available for

Compression-Ratio	Speed, r.p.m.	Power at 29.9 In. of Mercury and 59 deg. Fahr., b.hp.	Brake Mean Effective Pressure at 29.9 In. of Mercury and 59 Deg. Fahr., lb. per sq. in.	Fuel-Consumption, lb. per b hp-hr.	Spark Advance for Both Plugs, deg.	Mean Blast-Velocity, m.p.h.	T_{H2}	T_{H3}	Temperature Back of Exhaust-Valve-Guide Boss T_p	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}	T_{12}	Temperature Position
							ALUMINUM							STEEL								
							Temperatures, Degrees Fahrenheit Above Air															
5.3	1700	42.2	138.2	0.600	23	87	362	416	403	288												L.H.
5.3	1700	42.2	138.2	0.565	23	87	414	467	457	304	304	286	238	228	218	205	180	169	162	171	180	L.H.
Mean Temperature of Walls, Ports, Head and Barrel, 247 Deg. Fahr. above Air or 297 Deg. Fahr. Actual											333	353	348	333	308	326	313	295	284	279	286	Back
											317	259	214	175	184	139	131	126	129	142	180	Front
Maximum Circumferential Temperature Difference											29	94	134	158	144	187	182	169	155	137	106	
5.3	1700	41.4	136.6	0.541	24	87	396	454	450	328												L.H.
5.3	1700	40.8	134.7	0.530	24	87	425	456	459	326												L.H.

Fig. 289.—Table Showing Output Temperature and Fuel Consumption of the Airco Cylinder.

flow. This area, from the diagram, is 32 square inches.

Quantity of Air Needed for Cooling.—The quantity of air used on the above basis is 1,700 cubic feet per minute and is the product of the blast velocity in feet per minute and the area available for flow in square feet. The number of cubic feet required per minute per brake horsepower is 36.7. The heat to be dissipated per minute per brake horsepower is 25 B.t.u., or a total of 1,160 B.t.u. per minute. The weight of air required per minute is 129 pounds. The mean temperature-rise of the air is equal to the total number of British thermal units dissipated per minute divided by the weight of air required times the specific heat at constant pressure, or $1{,}160 \div (129 \times 0.24) = 37$ degrees Fahrenheit.

The cylinder used in this example is, of course, a highly efficient design, running under nearly ideal conditions. In fact, few air-cooled cylinders of such size have given better performance at a similar speed. The cylinder was running at continuous full-throttle, and even under these circumstances a considerable reduction in the blast velocity could have been made without materially affecting the performance or the temperature. Under conditions of intermittent full-throttle operation or the equivalent of progressively throttling that approximates the condition of a steady climb by aircraft, a considerable reduction of air supply could be made with safety.

Owing to the fact that the cylinder was operating in a free blast, much of the air supplied was not actually used for cooling, due to it striking the front of the cylinder and being deflected sideways and thus not passing through the fins at the cylinder sides. This may be an objection to the method used for estimating the air supply. Single cylinders have shown approximately the same performance when cowled so that all the air supplied was effective for cooling, as when running in a free blast of equal mean velocity. In cowled Vee engines, where the cooling air is supplied to the Vee and flows out sideways between the cylinders, all the air available for cooling has to pass through the fins and some of these engines have given remarkably good results, especially the Liberty air-cooled and the Wright Vee 1,456 shown at Fig. 231 in a preceding chapter.

Effect of Mixture Strength on Cooling.—A mixture-strength of twelve or thirteen to one by weight, when used on the composite aluminum and steel aircraft cylinder, usually results in a fuel consumption of approximately 0.55 pounds per brake horsepower-hour with aviation gasoline or aviation gasoline and benzol. This, however, only applies at the most suitable compression-ratio. If the ratio be too high, fuel-consumptions have to be increased to eliminate detonation. That a fuel consumption of approximately 0.55 pounds per brake horsepower-hour usually produces the maximum power and the maximum cylinder-temperature is confirmed by tests made by Mr. Heron. The maximum exhaust-valve temperature is usually produced by approximately the optimum mixture-strength of 15.2 to 1. The use of such mixtures is, however, impossible on anything but a first-class design. This question is dealt with later under the subject of exhaust-valve cooling. The specific figures quoted for the mixture-strength will doubtless be productive of criticism, since mixture-strength figures are tied up in the vexed question of volumetric efficiency. On the subject of the volumetric efficiency of cylinders of such outputs as are quoted in this

paper, no two investigators seem to be in agreement. The figures in which Mr. Heron has the most cause for confidence are therefore used.

The minimum fuel-consumption that an air-cooled cylinder will run on is generally a measure of its soundness. In this respect a rather curious difference is noticeable between efficient and unsound design. A poor design will usually work over a wide range of fuel-consumption, say from 0.7 to over 1.0 pounds per brake horsepower-hour, without much variation in the power output, whereas an efficient design will generally show a drop in the power output if the fuel-consumption at maximum load of approximately 0.55 pounds be increased by about fifteen per cent, the power progressively decreasing with a further increase of the mixture-strength. The reduction in the cylinder-wall temperature obtained when the mixture-strength is enriched beyond twelve or thirteen to one may be attributed to the increase of internal cooling by the evaporation of liquid-fuel particles, and to the reduction of both the rate of flame propagation and the flame temperature.

Effect of Air Blast Direction.—The relative importance of blast direction for fairly large cylinders is indicated by Table appended which gives figures of tests on an Airco cylinder with the blast on the exhaust and on the inlet sides of the cylinder, the temperature positions being shown in Fig. 287. The tabulation at Fig. 289, reproduced from the *S. A. E. Journal* and prepared by Mr. Heron gives the output, temperature and fuel consumption of the Airco cylinder, the various stations on the cylinder where temperature was measured being clearly shown at Fig. 287. For example, TH3 is a point on the aluminum head between the valve ports.

The circumferential temperature-differences at T_2, around the cylindrical portion of the combustion-chamber, with a change of the blast direction are noticeable. With exhaust-side blast the difference is 33 degrees Farenheit, and with inlet-side blast it is 301 degrees Fahrenheit. The increase with inlet-side blast of 197 degrees Fahrenheit in temperature at TH_3, the hottest point of the cylinder, is sufficient in itself to show the marked effects of blast direction. With inlet-side blast the fuel-consumption, although only slightly higher at the maximum-load mixture, could not be reduced as much as with exhaust-side blast. The minimum fuel-consumption with the inlet-side blast was nine per cent higher than that obtainable with the blast on the exhaust side.

Air-Cooled Cylinder Form.—The power output for a given size and speed is controlled largely by combustion-chamber design, size of valves, size and shape of ports, and cooling of the cylinder. It has been the experience of nearly all experimenters that the nearer one can approach to a spherical combustion-chamber, the better will be the results obtained. One of the most important considerations in securing maximum power output is the size and shape of the intake and exhaust ports. The internal shape of the ports must be such as to offer the minimum resistance to the flow of the gases in and out of the cylinders, while their external shape must be such as to accommodate reasonable forms of intake and exhaust manifolds, and to offer the least possible interference to the flow of the cooling air around the cylinder head. In order to offer the least possible restriction to

extensive research programs. The size and spacing of fins is largely controlled by the conductivity of the material used. However, for most materials, the scientifically correct finning calls for fins spaced so closely together and of such thin section that the manufacturing difficulties become prohibitive. Another consideration is that the fins must be sufficiently thick and strong to avoid damage in handling. A typical air-cooled cylinder of modern design is shown at Fig. 283, this being a composite structure.

Cooling Fin Dimensions.—The finning to be used, therefore, is the result of a compromise between these factors. It has been found that for aluminum or cast-iron cylinders, a very good design from all points of view is to make the fins one inch long with a thickness of 1/8 of an inch at the root and 1/16 of an inch at the tip, with a spacing of 3/8 of an inch between fins. For steel, the fins may be half as thick and spaced as closely as 1/4 of an inch. Fins should not be too closely spaced, however, or they will radiate heat from one to the other instead of to the air blast. Mr. C. B. Dicksee, of the Automotive Engineering Department of the Westinghouse Electric and Manufacturing Company writes in *Automotive Industries* on the subject of determining the cooling capacity of fins on crankcase or cylinder of an engine. He states that many attempts have been made and are still being made to take advantage of the high thermal conductivity of copper in the cooling of air-cooled engines. The conductivity of copper is over eight times that of cast iron or steel and it is only natural for designers to assume that the substitution of copper for the materials more commonly employed will materially improve the cooling of the engine.

A cooling fin has two functions to perform: First, to provide a surface from which heat may be removed by the air flowing over it; second, to provide a path along which heat may flow to the surface. The final result obtained is dependent upon the manner in which both of these functions are performed. It is obviously absurd to provide a large amount of surface if means are lacking for getting the heat to that surface, or to fit a fin of ample conductivity without giving it a corresponding amount of surface. In other words, for a fin to have the maximum efficiency the two functions must be in correct relation with each other.

Law of Heat Radiation.—The rate at which heat is removed from a body by air flowing over it is a direct function of the difference in temperature existing between the two It is also a function of the velocity of the air over the surface. From this it follows that the efficiency of a fin will depend upon its mean temperature measured above that of the air; while the ratio between this mean temperature and that (also measured above air) of the parent body from which the fin springs, will be a direct measure of the effectiveness of the fin relative to the surface of the parent body. This ratio may be conveniently expressed by the following equation:

$$f = T_m/T_1,$$

where f = the effectiveness of the fin, T_m = the mean temperature of the surface of the fin and T_1 = the temperature of the parent body, both measured above air.

Heat Flow in Fins.—The temperature of the surface of the fin will fall from a value at its root equal to that of the parent body to some lower value

at the extreme tip. The difference in temperature between any two points (*a* and *b*, Fig. 291) across the surface of the fin will be directly proportional to the distance between them and the rate of heat flow, and inversely proportional to the thermal conductivity of the material from which the fin is made and to the area of the path along which the heat is flowing.

This fundamental relationship may be expressed by the following equation:

$$t = \frac{Q \times D}{K_t \times a} \qquad (1)$$

where t = temperature difference between the two points considered.
Q = rate of heat flow in watts.
D = distance between the points in inches.
K_t = thermal conductivity of material in watts/square inches in degree Centigrade.
a = area of path along which the heat is flowing, in square inches.

It will be well here to state that to simplify the analysis the heat is

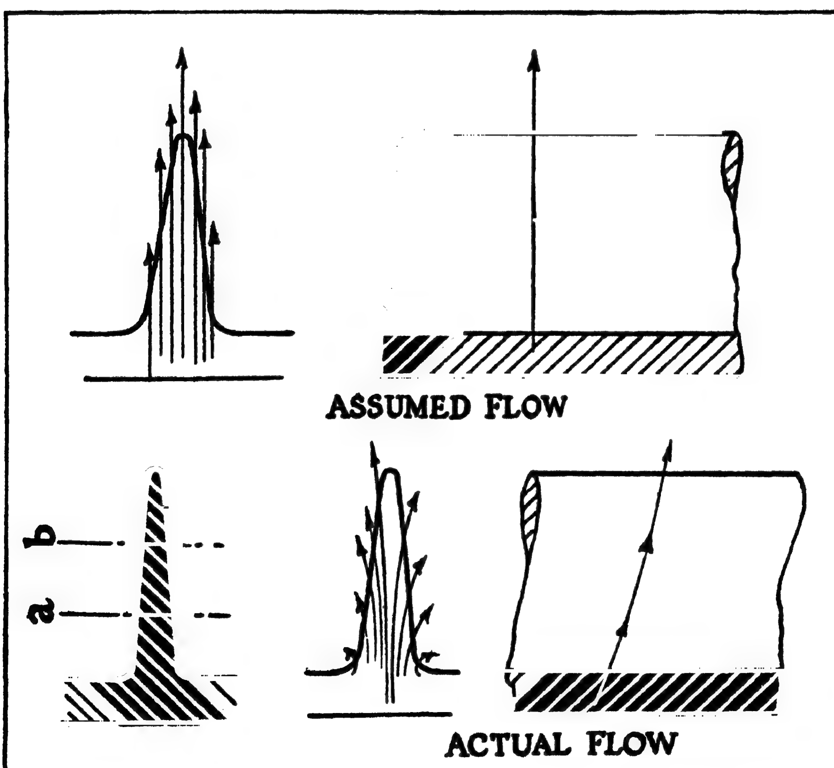


Fig. 291.—Diagram Showing Actual and Assumed Flow in Taper Section Heat Radiating Flange.

assumed to flow directly from root to tip of fin in a direction parallel to the axis of the fin. *This is not strictly true, as actually the heat flows from the* inner part of the fin toward the surface and in most cases in practice there will also be a flow in the third direction along the length of the fin, due to differences in temperature at different points on the surface of the parent body. The illustration will make this point somewhat clearer. In the case of a cooling fin heat is lost from all points across the surface, so that the rate of heat flow past the point *a* (Fig. 291) will be greater than that past the point *b*, which is more remote from the parent body. As a consequence, the temperature gradient across the fin tends to decrease toward the tip. Where fins are either cast or machined from the solid metal they are usually tapered toward the tip, so that the area of path provided for the heat decreases as the distance from the parent body increases. This tends to increase the temperature gradient, but, the taper is never such as to offset entirely the effect of the reduction in the quantity of heat flowing. It is possible to design a fin such that the temperature will decrease at a constant rate from root to tip, but such a shape is impossible for practical reasons.

When copper fins were employed they were usually made from sheet metal attached by some means to a steel or cast-iron cylinder barrel. It has also become accepted practice to cast into iron barrels fins made from sheet or strip steel. In such cases the fins will have a rectangular cross section, and it is chiefly with such fins that it is proposed to deal in these notes.

Rectangular Fins.—The mean temperature of a fin of rectangular cross section and finite height may be calculated from the following equation, which is derived from the fundamental equation quoted above:

$$T_m = \frac{T_1 \tanh b\,h}{b\,h} \qquad (2)$$

where T_m = mean temperature of fin above air, degree Centigrade.

T_1 = temperature of parent body above air, degree Centigrade.

h = height of fin, inches.

$$b = \sqrt{\frac{K_v \times s}{K_t \times a}}$$

K_v = rate of heat exchange between surface and air, watts/square inch/degree Centigrade difference.

s = perimeter of fin in inches.

a = area of path provided for heat flow, in square inches.

In general it will be found convenient to deal with a strip of fin one inch long. In this case, s becomes two and a becomes equal (numerically) to the thickness of the material forming the fin.

As already stated,

$$f = T_m/T_1,$$

$$\text{hence } f = \frac{\tanh b\,h}{b\,h} \qquad (3)$$

A few moments' reflection will show that the expression b is capable of a further slight simplication. The two factors composing the denomina-

tor are those which determine the value of the fin as a conductor of heat, just as the value of an electric conductor is determined by its cross section and the resistivity of the material. The product of these two factors, which may conveniently be called the "conductivity factor" and designated by the symbol C, forms a valuable basis of comparison. The value of K_t changes with the material, while a is solely dependent upon its thickness. Hence, by suitably choosing the thickness, the value of C may be kept constant, and also the value of f, whatever the material employed.

Actually, a change in thickness will result in a change in the area presented by, and therefore the heat dissipated from, the end of the fin, but in most cases this will be so small as to be neglible. For a given height of fin, a series of curves may therefore be drawn showing the variation of effectiveness with the conductivity factor at different rates of heat dissipation, and this series may then be used to determine the effectiveness of fins of that height in any material.

Such a series was prepared by Mr. Dicksee which was drawn for fins 1⅛ inches high. From this it was pointed out that at first the value of the effectiveness factor increases at a rapid rate, but that later the rate falls off till there is only a relatively small increase for a material increase in the conductivity factor. This shows that there is a limit to the value of the conductivity factor beyond which it will not pay to go. More particularly is this the case at the lower rates of heat exchange which are obtained at low air velocities. Low rates of exchange give a well defined knee in the curve; at the higher rates this is not so well defined, and the economic limit is therefore not quite so definite, but may none the less be fairly closely determined.

A line drawn crossing the group of curves indicates approximately the limit for the 1⅛ inch fin. The more clearly to illustrate what this means, the following table, which gives the economic limits of C for the 1⅛ inch fins at several rates of transfer, is appended. This shows the thickness which is represented by these values in several different materials.

Rate of exchange, Watts/sq. in./C, (Kv)	.02	.04	.06	.08	.10	.12
Economic unit of Conductivity Factor	.09	.125	.145	.165	.175	.190

Material	Thermal Cond't'y (Kt)	Thickness (C/Kt), Inches					
Copper	9.0	.010	.014	.016	.018	.019	.021
Aluminum	5.0	.018	.025	.029	.032	.034	.038
Brass	3.0	.039	.042	.048	.055	.058	.063
Steel or Cast Iron	1.1	.082	.114	.132	.150	.159	.173

Some of the thicknesses given in the table are obviously out of the question from practical considerations, and this is particularly so in the case of copper, where the economic thickness is only 0.021 inches for this height with a rate of transfer as high as 0.12 watts/square inch/degrees Centigrade. The determination of rate of heat transfer in Mr. Dicksee's test was measured electrically which explains the use of a fraction of a watt as a unit of measurement.

It must be remembered, however, that the smaller the quantity of air used the greater will be its increase in temperature, so that while the temperature difference between the air and cylinder may remain practically unchanged, the mean temperature of the cylinder has actually increased. While it is temperature difference between air and metal that governs the cooling, it is true temperature which governs engine operation. A further point is that the rate of dissipation does not remain constant in the case of an automobile engine. Actually the rate of dissipation varies as a function of the engine r.p.m. The pressure produced by the blower or fan, or the slipstream of the propeller varies as the square of the r.p.m. and the velocity of flow of air through any system varies as the square root of the pressure, hence the velocity of the air varies directly as the r.p.m. The rate of dissipation therefore bears the same relation to the r.p.m. as it does to the air velocity. The change in rate of dissipation with speed must consequently be taken into account when arriving at the actual proportions of the fin which are to be adopted. It will be found that for any particular fin the final cooling effect as given by the product $(A \times K_v)$ will continue to increase with an increase of K_v despite the decrease of effectiveness.

Rates of Heat Dissipation.—These notes would be incomplete if some mention were not made of the rate of dissipation which is to be expected at a given air velocity. The whole question of dissipation is a very involved one, and various observers do not agree as to absolute values. In a previous discussion Mr. Dicksee stated that the rate of dissipation was given by the equation $K_v = 0.0145\ V^{.89}$, where K_v = rate of dissipation in watts/square inch/degree Centigrade diff., and V = velocity of air in thousandths of feet per minute. These values, which were obtained from some experiments in which the air flowed in a fairly smooth stream, would appear to be somewhat too low for engine work. In most cases in actual practice the air stream will be in a decidedly turbulent state and as a result the rate of exchange will be increased.

Uncertainty as to the true rate of heat transfer, however, will not in the least detract from the value of the analysis of any proposed arrangements of fins, as results will have relative values which are directly comparable with each other, and the advantage or otherwise of any one particular arrangement over another may be readily determined. In the foregoing reference was had practically entirely to fins having a rectangular cross section. The difference in results produced by tapering the fins is, however, surprisingly small. This is particularly so in the case of fins of thicker sections such as are suitable for casting or machining from solid metal. The actual error from treating a tapered fin of moderate section as one of rectangular section equal in thickness to that at the root of the tapered one will be relatively small. Tapered fins do not lend themselves to solution by a simple equation such as that given for rectangular ones.

Mr. Dicksee states that it would appear that the advantages to be gained by the substitution of copper for steel or cast iron will not be by any means as great as would be expected from a consideration of their relative conductivities alone. To reap the greatest benefit from the high

conductivity of copper, the material should be applied in the form of numerous thin fins. The problem of securing such fins commercially and economically to a steel or cast-iron barrel still awaits a satisfactory solution. The mere substitution of copper in the same dimensions will not result in any great improvement, though in some instances it may be beneficial.

Circumferential Finning Best.—The circumferential fin, on the whole, appears to have the majority of the advantages according Mr. S. D. Heron, for aviation engine cylinders. It gives a considerable stiffening effect to the cylinder that is of advantage in resisting distortion under temperature, explosion and bending stresses. Circumferential finning in general simplifies the problem of best applying the air-supply to the cylinders. A combination of circumferential fins for the barrel and the cylindrical portion of the head with circumferential and axial fins or axial fins alone for the crown is used at times and is practically essential for four-valve roof-head types. In the foundry a cylinder with circumferential finning is a decidedly superior production proposition to one completely finned with radial axial fins. The pivotal point in the design with a cast air-cooled cylinder or head is really foundry production. It is useless to produce a design that will not mould readily or that calls for complicated cor'ng.

The use of axial radial fins makes it extremely difficult to provide any effective finning of the cylinder crown, and most types of such cylinders exhibit a singular bareness as regards such finning. It is, of course, easy to design a crown covered with fins, but to produce a fin layout for it that can be cast readily and that will give an efficient air-flow through the fin spaces is another matter. In automobiles, where blower cooling is used and air from a pressure blower is directed by a conduit to the top of the cylinders, as in the Franklin, axial radial fins are absolutely necessary and heat absorbing qualities are secured by using plenty of metal in the head to make up for lack of finning.

The finning and cooling of the cylinder-head is much more important than that of the barrel. Most axial-fin designs rely upon the barrel finning to cool the cylinder crown, a very thick crown being used to conduct the heat from the center of the head to the barrel. Such practice is not sound, as it places a double duty upon the finning of the combustion-chamber sides and, further, the heat is not dissipated from the crown at the point of its reception. Attempts to use such a construction, although producing passable results in cylinders of small capacity, have, in Europe at any rate, invariably given trouble when applied to cylinders of over 100 cubic inch capacity. Reliable figures have yet to show that such a design in any size can compare in performance with cylinders in which an attempt is made to dissipate the heat at its point of reception.

Overheating of the lee side of a circumferentially finned cylinder does not occur in a cylinder of correct design. The view generally accepted in this country that overheating of the lee side of a circumferentially finned cylinder is inevitable has arisen mainly from the performance of very light steel aircraft-engine cylinders, mostly of European origin, and all of poor design. Thin cast-iron cylinders with the air-supply unsuitably arranged have also helped to confirm the fears of lee-side overheating.

For aircraft cylinder-heads and all cast-iron or semi-steel cylinders the cast fin appears to be the most logical. For automobile- or motorcycle-engine cylinders the cast fin is undoubtedly the cheapest and the power output obtained does not justify the use of fins produced by machining or casting-in except on highly refined engines of costly cars such as the Franklin. On aircraft-engine cylinder barrels machined fins can be used on account of the reduced weight, but this calls for expensive construction because the tubing or forging used must have a wall thickness great enough to permit machining fins of the proper height which means that considerable metal must be removed from the rough cylinder and wasted.

Types of Air-Cooled Cylinder Head.—In general the work on which the experience of Mr. Heron is based shows that the flat head does not compare favorably, as regards either power output or cooling efficiency, with the spherical or roof types in air-cooled cylinders. The reasons for this are plain. In the first place, the flat head renders it difficult to maintain the required air-spaces between the valve ports and the necessary metal sections between the valve-seats without reducing the valve sizes excessively. Large air-spaces between the inlet- and exhaust-valve ports are desirable with three- or four-valve designs employing a pair of exhaust-valves, and it is essential to have a minimum air-space of 3/4 inch between the adjacent exhaust-port walls in such types. Further, with aluminum heads, it is necessary to maintain approximately 5/8 inch as a minimum section of aluminum between adjacent exhaust-valve seat-inserts; otherwise overheating, distortion and cracking will be apt to occur. It is not wise to reduce this section much between pairs of inlet seats or between adjacent exhaust and inlet seat-inserts.

With a flat head the walls of adjacent ports do not diverge as much from each other as with roof or spherical heads and thus the area of the total air-space between the ports is less. The included angle between the entrance and the exit of the ports is nearly always smaller with flat heads than with roof or spherical types. With the latter it is nearly always possible to maintain the included angle at more than 90 degrees, which is of considerable importance, at least for the exhaust-ports, as there is thus less gas friction and therefore less heat given to the port walls. With an exhaust-port included angle of less than 90 degrees it is easier to avoid choking the port while maintaining good valve-stem cooling by a heavy valve-guide boss and, further, the port walls diverge more sharply from the combustion-chamber wall. Even if the included angle of the port be 90 degrees, a greater divergence between the port walls and the cylinder crown is obtained with the spherical or roof type heads, which is of importance for cooling, as the air is thus more uniformly in contact with the metal surrounding the valve-seat. For efficient cooling it is desirable that the fins be, as nearly as possible, normal to the surfaces to be cooled.

The side-valve cylinder so suitable for motorcycle engines has marked disadvantages in comparison with the overhead type for aircraft use. Combustion-chamber areas are relatively greater, equally efficient finning cannot be obtained, the temperature distribution is perforce much more uneven and distortion is much more likely to occur. Distortion is exceed-

ingly difficult to avoid in side-valve cylinders. The inlet and exhaust ports usually distort the barrel, with the result that piston contact is poor and the ring fit uneven. The valve-seats rarely remain true when hot, with the result that grinding-in the valves with the cylinder hot is resorted to at times.

The side-valve design of either the L or T form of necessity departs from the symmetry so desirable and in fact absolutely necessary for an air-cooled cylinder. For equal class of design and development, the mean effective pressure of a side-valve cylinder is usually twenty per cent less than that obtainable from an overhead-valve type, with continuous full-throttle operation.

How Auto and Aviation Practice Differ.—L-head engines, which at present are most commonly used on automobile engines present the advantage of requiring the minimum number of operating parts. A single camshaft is used to operate both the inlet and exhaust valves and, as far as the camshaft is concerned, a single pair of gears or sprockets connecting it with the crankshaft suffices. This same advantage is secured by an overhead valve engine but longer push rods and rocker arms are called for unless the camshaft is also placed overhead. The L-head engine is also superior to the T-head in the matter of combustion-chamber exposed to the water-jacketed wall. This fact is perfectly obvious and we have been able to obtain thermal efficiencies with small L-head engines which are fully as good as those obtained with any other valve locations in passenger-car engines though the requirements of aviation engines call for the I-head arrangement. The volumetric efficiency possibilities of an L-head are slightly inferior to those of a T-head engine but, in practice, this difference is more than offset as a rule by the advantage gained in thermal efficiency. Incidentally the L-head engine, due to the minimum of moving parts, is capable of being made more silent in operation than any of the other types of poppet-valve engine, and silent operation is one of the most highly-prized qualifications in the automobile engine, though not so important in aviation engines where maximum efficiency is desired and lowest possible weight-horsepower ratio rather than great quietness is sought.

The overwhelming advantage of an I-head engine is that it presents a minimum amount of cooled combustion-chamber wall and therefore comes nearer to forming the ideal combustion-chamber, which should be hemispherical since the largest capacity for the least area of wall is contained in a sphere as was fully outlined in our consideration of thermodynamics. We thus find that very high thermal efficiencies can be obtained with I-head engines and, owing to the ideal combustion-chamber, very high compressions can be carried even in large-bore cylinders without preignition and detonation, provided proper material and thickness of wall are used. Hence I-head engines are universally used for aircraft work. The problem of operating the valves quietly has never been satisfactorily solved. The camshaft can be located adjacent to the crankshaft and the valves operated by suitable push rods and rocker arms as in the Cirrus engine or the radial cylinder air-cooled types or it can be placed next to the valves, acting either directly against the stems as in the Fiat, Hispano-Suiza, Curtiss and Packard engines or by rocker arms as in the Liberty, Lorraine and others, as previously outlined.

Large Cylinders Air Cooled.—There is a very general tendency to assume that air cooling is suitable only for small cylinders. It may be well to quote the performance of the R.A.E. 19T cylinder, illustrated in Fig. 292, to show the extent to which the air cooling of large cylinders has gone. This cylinder is of eight inch bore and ten inch stroke, or of 502.6 cubic inches capacity, and has developed 129 brake horsepower, or 119.5 pounds per square inch brake mean effective pressure at 1,700 r.p.m., with a fuel consumption of 0.51 pounds per brake horsepower-hour. The cylinder was built to determine the possible limits of air cooling and, although it has at present little practical application, it has at least demonstrated that successful air cooling is not limited to 50 brake horsepower per cylinder. It will be observed that this construction is one in which the aluminum alloy head is cast on the cylinder barrel and that four valves are used in the spherical head.

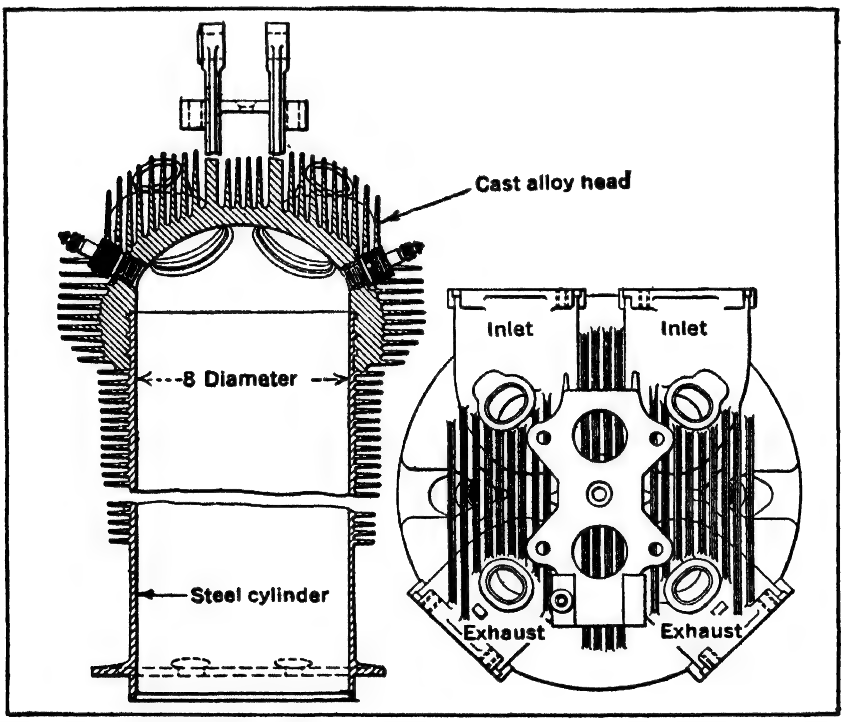


Fig. 292.—Plan View and Sectional Elevation of the 19-T, Eight Inch Bore by Ten Inch Stroke Aircraft Engine Cylinder, with Cast Aluminum Head, Developed by the British Royal Aircraft Establishment.

Fragility is a disadvantage of air cooling as at present developed, the fins of a cast-iron automobile-engine cylinder of even the sturdiest design being relatively delicate in comparison with the cast-iron water-cooled block of an engine for similar use. There is little to choose, however, between the fragility of aircraft types if the composite aluminum and steel air-cooled cylinder be compared with the built-up all-steel water-cooled

cylinder. In fact, the former, if anything, has the advantage, as detonation, preignition or vibration will often crack water-jackets but does not result in fin damage. It is not possible at present to produce air-cooled engines having the delightfully clean outline of water-cooled engines because the nature of the cooling medium calls for numerous projecting fins whereas water flow is restrained and confined properly by smooth wall jackets, the heat being taken from the water by radiators independent of the engine.

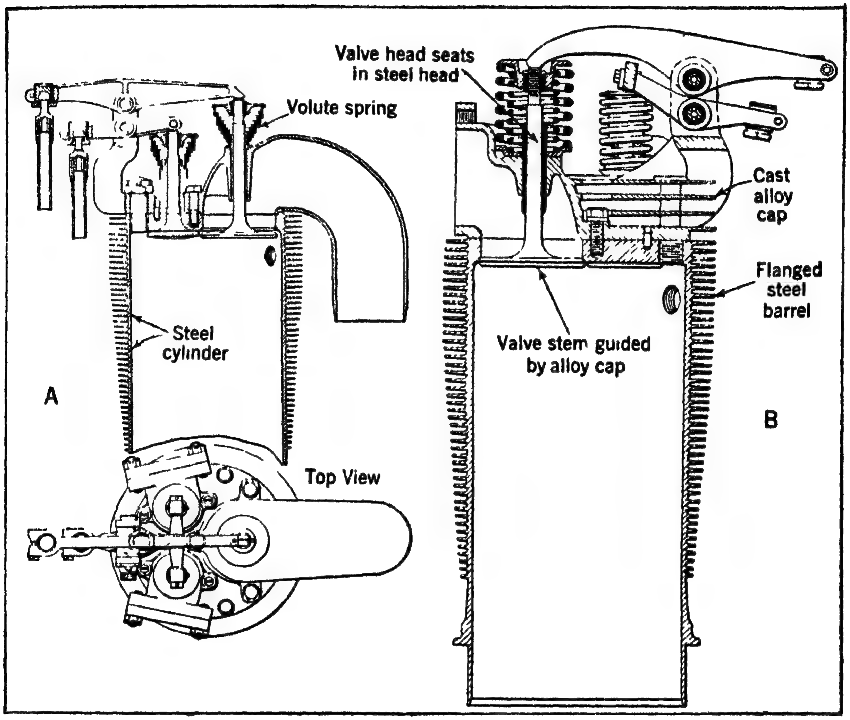


Fig. 293.—View at A Shows an All-Steel Five and One-Half by Six and One-Half Inch Air-Cooled Engine Cylinder Having Bolted on Valve Cages. At B a Sectional Elevation of a Steel Cylinder with Aluminum Cap Pieces, Carrying Valve Stem Guide and Rocker Arm Assembly is Shown.

An air-cooled cylinder has its heat radiator directly attached and formed integrally, the heat flows to it quickly and is removed by the energetic air blast in the most direct manner possible.

Spherical and Roof Heads.—The discharge of an inclined valve in a spherical or roof head appears to be much less disturbed and more likely to promote and maintain turbulence than that of a similar valve in a flat head. The flat head is more liable to deflect under explosion pressure and, in practice, breathing of flat heads is by no means unknown. The truly hemispherical head is the ideal form as regards minimum stress due to explosion pressure. For two-valve designs the spherical head appears to be the most suitable. For four-valve heads the spherical head is undoubtedly the most efficient, but, except for very special cases, its use does not appear to be

justified, owing to the manifest difficulties of valve operation and also to the fact that excellent results are obtainable from a suitably designed roof head. When, however, very large cylinders are required, or medium-size designs, either of large bore-stroke ratio or to run at such high speeds that sufficient valving in conjunction with good cooling cannot be obtained with a roof head, there is little question that the spherical head is the most suitable design. An excellent example of four-valve spherical-head design is seen in the R.A.E. 19T cylinder, Fig. 292.

Experience to date indicates that for aircraft-engine cylinders of up to about 170 cubic inches capacity and with bore-stroke ratios not exceeding 1.25 to 1, designed for normal speeds of up to 1,800 r.p.m., the roof head will produce the best all-around results. The design of an efficient roof-head cylinder is not easy as slight faults in detail are liable to result in poor cooling.

Investigation indicates that for aircraft-engine cylinders the two-valve type is best for capacities of up to 100 cubic inches per cylinder and that beyond this size the four-valve type is superior; although two-valve cylinders of up to 160 cubic inches capacity have been built and given excellent results. The large two-valve type, however, is not as a rule suitable for speeds in excess of 1,700 r.p.m., usually involving very high gas-velocities through the valves and not giving nearly such a high performance as the best class of four-valve cylinders. Experience has shown that unless a four-valve air-cooled cylinder of up to 160 cubic inches capacity is very carefully designed, the performance will be inferior to that of a good two-valve design of similar size. Large three- and four-valve air-cooled cylinders of the flat-head type have proved to be much less efficient than the two-valve spherical-head type, though they have been widely used in water-cooled cylinders.

Composite Cylinder Constructions.—Air-cooled cylinders have been made in a variety of forms. The head and cylinder cast in iron or semi-steel in one integral casting is the simplest and cheapest. The form in which the head and radiating fins are cast in alloy and a steel sleeve shrunk in is seldom used. The use of an aluminum alloy head cast in place on a steel or cast-iron barrel has been discontinued in favor of the screwed on cast alloy head, having its own fins and a cylinder barrel of steel with cooling fins machined thereon threaded into the head.

Alloy Head Cast on Steel Barrel.—Some of the pros and cons of the various types of overhead-valve composite aluminum and steel aircraft-engine cylinder construction are considered below as summarized by Mr. S. D. Heron. The head cast on an integral-fin steel barrel such as shown at Fig. 292 possesses the advantages that

(1) No machine-shop operations are required to fit the head to the barrel
(2) It produces probably the lightest possible design

The disadvantages of this type are:

(1) Very high casting stresses are set up in the head, which are liable to cause failure in service and cannot safely be removed by annealing, as this would largely remove the shrink of the head on the barrel

(2) The alloy in the head casting is in a state of instability owing to the lack of annealing
(3) It is next to impossible to inspect the condition of the contact of the head with the barrel
(4) Foundry difficulties are numerous; although in times of peace they can be overcome, the great care and supervision required are unsuitable for the stress of war time. The finished casting represents but a very small percentage of the weight of the metal melted to produce it
(5) Barrel production has to start before that of the heads and the complete cylinders
(6) Removal of an unsound head casting from a barrel may scrap the latter
(7) Damaged heads or barrels cannot be removed from one another in the field and new parts fitted to the sound portion
(8) The finned steel barrel is somewhat expensive to produce and wastes a lot of material
(9) Fitting of expanded or shrunk-in valve-seats is a poor production job with tilted valves
(10) If a roof-type head be adopted, machining is difficult

Bolted-On Separable Heads.—The advantages of the bolted-on head are:
(1) Ease of production and assembly
(2) Ease of replacement of damaged parts

This type of head possesses the disadvantages of
(1) Difficulty of maintaining pressure tightness at the joint between the head and the barrel; if any flame gets through, the head is burned out
(2) Axial heat-flow from the head to the barrel is practically non-existent. The cooling efficiency is reduced and as a result head temperatures and fuel-consumptions are increased

Cast Cylinder of Alloy with Liner.—The type of cylinder construction in which the head and the jacket are an integral casting with a shrunk-in liner as shown at Fig. 290 A possesses the advantages that
(1) The liner can be produced from a thin tube and is cheap to machine
(2) The jacket and the head is a good production job in foundry

The following objections are raised to this design of cylinder
(1) The whole explosion load is taken through the jacket, the load being transmitted to four or more bolts. This arrangement requires a heavy jacket and even then is not satisfactory, for aluminum at the temperatures obtained has not proved suitable for the resulting heavy localized stress at the bolt-bosses. The cast-on or screwed-on head types transmit the explosion load from the head to the barrel and thence to the holding-down bolts, with practically uniform stress in the circumference of the head metal
(2) Unless a heavy liner is used, it is difficult to prevent distortion and lack of contact between the liner and the jacket. It is very difficult to prevent oil from leaking between the liner and the jacket, which causes a lack of contact owing to carbonization. Oil leakage, with the resultant carbonization, is much less serious with a screwed-on

head than with a shrunk liner, as the fitting surface on a shrunk liner can be covered completely with carbon, whereas the screwed-on head carbonizes only on the unloaded faces of the threads

(3) To eliminate oil leakage as far as possible the joint of the liner with the jacket should be made at the liner top. Owing to the

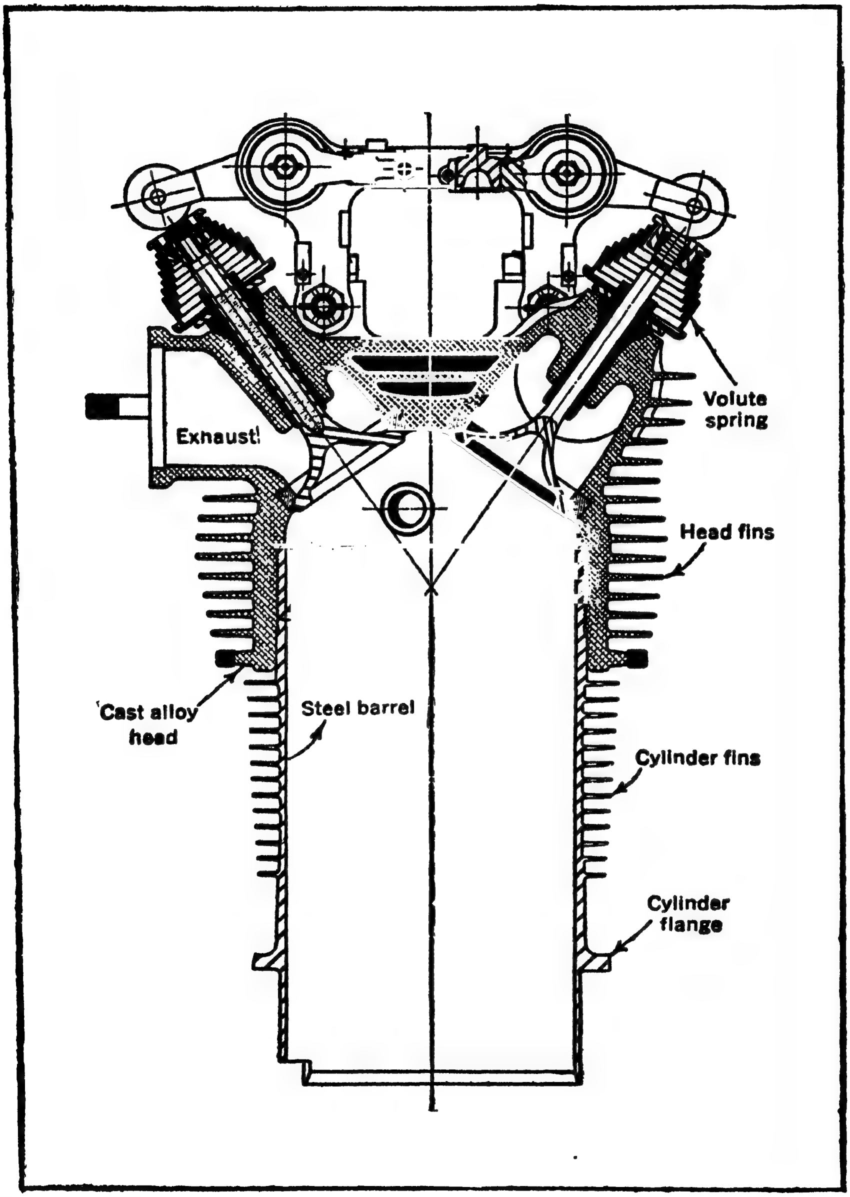


Fig. 294.—Type J Air-Cooled Engine Cylinder which will Stand a 100-Hour Full Throttle Test at 1,800 R.P.M. without Measurable Wear. Note Salt-Cooled Exhaust Valve Stem, and the Use of Volute Valve Springs.

difference in coefficients of expansion of the materials of the liner and the jacket, much ingenuity is required to determine the correct position in which to attach the holding-down bosses to the jacket so that the bolts may be under constant tension. Trouble occurs owing to the jacket deflecting at the holding-down bosses and locally distorting the liner

(4) The increase of weight with this construction is greater than that of the cast-on head and jacket type, owing to all explosion load being transmitted throughout the length of the jacket

Steel Barrel with Alloy Cap.—The advantages of the steel barrel with an aluminum cap as shown at Fig. 293 B are:

(1) The cap is easily fitted
(2) Pressure tightness is not dependent on fit of cap

Opposed to the foregoing advantages are:

(1) The closed-end barrel is more expensive to machine than the open-end type used with a cast-on or screwed-on head design
(2) The state of contact of the head is unknown after a little service, and this contact is difficult or impossible to maintain. Flame-blowing between the cap and the top of the barrel has been known to occur
(3) Can only be used for flat heads
(4) Gives a poor performance in comparison with cast-on or screwed-on spherical or roof heads
(5) The valves are seated in the steel head and are guided in the aluminum cap. The difference in the coefficients of expansion of the two parts is liable to cause misalignment of the valves on their seats, contributing to valve burning, warpage and leakage generally characteristic of this construction
(6) Owing to the relative expansion of the cap and the barrel due to the difference in the coefficients of expansion of the materials, either sliding must occur between the two contact faces or heavy stresses be set up in the cap and the bolts attaching it to the barrel

Sometimes, all steel cylinders are used having bolted-on members to support the valve-stem guides and to provide outlet and inlet port passages for the gas. Such a construction is shown at Fig. 293 A and most of the disadvantages mentioned for the steel cylinder and aluminum cap construction shown at Fig. 293 B and previously enumerated in detail apply to this construction as well as it is the same generic type as the other.

Material for Air-Cooled Cylinders.—Considering the material of the cylinder from the point of view of cooling alone, we require:

(1) A material having a high rate of heat transfer from its surface to air.
(2) A material having rapid internal conductivity, so that the heat will flow readily from the hotter to the cooler parts of the cylinder, thus avoiding local overheating.

No definite data on the rate of heat transfer to air is available, but from experience it appears that copper, aluminum, steel and cast iron possess this property to a high degree, as shown by the research of Mr. Dicksee.

Of the metals having high heat conductivity, aluminum alloy stands pre-eminent on account of its added advantage of light weight, although copper and its alloys have been proposed and used. Copper transmits heat best, then comes aluminum alloy, then cast iron and steel. The question of cylinder materials, however, is largely governed by considerations other than those of cooling, the important one being the resistance to wearing due to piston reciprocation.

Cast iron will fulfill all of the functions in a fairly satisfactory manner, but its weight is generally prohibitive for military engines. For commercial engines cast-iron cylinders may often be used to advantage. Cast aluminum alloy is satisfactory as a containing member and can be cast in intricate shapes. It will also stand a fair degree of stressing, and is a fairly good bearing material. However, where high localized stresses are encountered, cast aluminum alloys are liable to failure by fatigue, and consequently have been found unsatisfactory for the hold-down flange, where the cylinder is bolted to the crankcase, unless very heavy sections are used.

In order to overcome these difficulties, a combination of materials has usually been resorted to. Steel makes an excellent bearing surface for the piston. It is now almost universal practice, where the cylinders are not entirely of cast iron to make the barrel of steel with an aluminum head.

FACTORS OF SAFETY, AIR-COOLED CYLINDER PARTS

Part	Material	Direct Stresses	Other Considerations	Recommended Factor of Safety Based on Ultimate Strength at Room Temperature
Cylinder Barrel at thinnest section	Steel	Tension due to pressure on cylinder head	Rapidly varying load Vibration Fairly high temperature	8
Hold down studs (Root diameter)	Steel	Tension due to pressure on cylinder head	Rapidly varying load Severe handling stresses	12
Sides of combustion chamber at thinnest section	Aluminum Alloy	Tension due to pressure on cylinder head	Rapidly varying load High temperature	10

Investigation has shown that a steel surface gives five to ten per cent greater heat-dissipation than either aluminum or copper. The dissipation of aluminum, however, is improved about ten per cent by coating with a glossy black enamel, the percentage of improvement varying with the nature of the enamel and the blast velocity. The effect of surface dissipation is of considerable importance and further investigation of the subject is to be made by the Bureau of Standards and the engineering division of the Army Air Corps.

Alloys for Cylinder Heads.—The principal alloy that has been used by the British for air-cooled cylinder construction is the Air Ministry 2-L-11 alloy, containing seven per cent of copper, one per cent of tin and 92 per cent of aluminum. This was developed after failure had attended efforts to use the aluminum-zinc group, employed so largely on the Continent for crankcases and similar parts. This alloy is both weak and soft when hot, but its bad qualities are, nevertheless, fairly well understood as a result of considerable experience. It sand-casts relatively well; the tin-content, which reduces the strength and hardness, also seems to reduce shrinkage and pin-holing.

An alloy, known as the Y-alloy, developed by Dr. Walter Rosenhain at the National Physical Laboratory, contains four per cent of copper, two per cent of nickel, 1½ per cent of magnesium and 92½ per cent aluminum. This maintains its strength and hardness at high temperature much better than the 2-L-11 alloy and has been used to a limited extent for cylinder castings. It appears to have great possibilities for air-cooled cylinder construction, principally on account of its hot strength and hardness, but much will depend on its ease of casting. For air-cooled seaplane engines it is possible that this alloy may prove distinctly advantageous, owing to its remarkable resistance to corrosion by sea water.

The silicon-copper-aluminum alloys appear to be one of the most promising groups yet tried for cylinder construction and at present the only unknown factor concerning the group is its conductivity. It seems that this group, which at present is under development by the engineering division and others, will mark a distinct forward step in alloys for air-cooled cylinders. Its freedom from cracking, porosity and shrinkage appear to go far toward eliminating the troubles of the commercial production of aluminum air-cooled cylinder castings. An important advantage of this alloy is that chills, which in production with jolt-ram moulding machines by unskilled labor are exceedingly objectionable, can be entirely eliminated.

Some particulars of the casting of silicon alloys for air-cooled cylinders have already been published, and it is hoped that the results of the further investigations of the materials section of the engineering division will be published in the near future. The silicon-copper alloy, containing in addition to 93 per cent aluminum, four per cent of silicon and three per cent of copper, possesses considerable ductility, which is of some practical importance as bent fins and the like made of it can be straightened without danger. Most of the silicon-alloy cylinders produced by the engineering division have been cast in this alloy and have proved somewhat difficult to machine. Threading has given most trouble in this respect, owing to the tearing of the metal and the wear of the tools. The machining properties are, however, likely to be much improved by the investigations now being conducted. The silicon-copper alloy presents an excellent example of the danger of judging alloys for cylinder construction by test-bar results alone. From the test-bar results little consideration of the silicon-copper alloys and less of the 2-L-11 alloy would be justified, though both have proved excellent. Test-bar results entirely fail to show casting properties, an aspect of marked importance for air-cooled cylinder production.

Alloys containing copper and manganese which have a constant or an increasing strength up to 500 degrees Fahrenheit, have been tried for air-cooled cylinder castings. Owing principally to casting difficulties they have been dropped in favor of more promising groups. The conductivity of alloys containing manganese is fully developed only after annealing. In some cases the increase of conductivity due to annealing amounts to as much as 40 per cent. Most alloys show a somewhat improved conductivity after annealing. Apart from this effect, annealing is very necessary to remove growth and casting strains. If cylinder-heads are not annealed, growth and distortion develop rapidly in the course of engine operation. With an unannealed screwed-on head the effect of growth is most serious; such a head will become loose on the cylinder barrel in ten to twenty hours running. Growth due to lack of annealing further results in valve-seat inserts, valve guides and cylinder studs becoming loose in the casting.

The physical properties of some aluminum alloys used for cylinder castings are shown in Table below.

PHYSICAL PROPERTIES OF ALUMINUM ALLOYS EMPLOYED FOR SAND-CAST CYLINDERS

Alloy	2-L-11[f]	Y[g]	Copper Silicon[h]
Composition	Cu 7%	Cu 4%	Si 4%
		Ni 2%	
	Sn 1%	Mg 1½%	Cu 3%
Tensile-Strength, lb. per sq. in.			
60 Deg. Fahr.	14,800	24,400	21,400
300 Deg. Fahr.	12,300	23,800	19,400
600 Deg. Fahr.	8,300	23,000	11,800
Brinell Hardness Number			
60 Deg. Fahr.	55	71	50
300 Deg. Fahr.	54	68	..
600 Deg. Fahr.	25	46	..

[f] See report on the Materials of Construction Used in Aircraft and Aircraft Engines, by Lieut.-Col. C. F. Jenkin, published by His Majesty's Stationery Office, London.

[g] See the 11th Report of the Alloys Research Committee of the Institution of Mechanical Engineers.

[h] From tests made by the materials section of the engineering division of the Air Service.

For cast-aluminum cylinders a fin having a length of from 1 to 1⅛ inches, a thickness at the root of ⅛ inch, a thickness at the tip of 1/16 inch and a ⅜ inch pitch, is fairly readily cast and is reasonably strong for handling. Fins of a somewhat decreased thickness and pitch can be cast, but are likely to lead to foundry difficulties and are easily damaged.

Cast Iron for Cylinders.—Cast iron at present is somewhat despised as a cylinder material. Nevertheless, efficient results are obtainable with this material. For commercial purposes where extreme lightness is not required excepting military aircraft engines, the use of any other material than cast iron for cylinder construction is not justified by the increased efficiency obtained. The use of any more expensive material than cast iron also involves more complicated and expensive construction. In the light of present experience that statement appears to hold good up to an output

of fifteen to twenty horsepower per cylinder at about 2,500 r.p.m.

A comparison of the weight of the cylinder-body proper, excluding the valves and the valve-gear, obtainable with heavy cast-iron construction and by the use of advanced aircraft-engine design, with the weight reduced to a minimum, may be of interest. An automobile engine cast-iron cylinder of massive proportions, with little liability to breakage that will give excellent cooling, will weigh two pounds per horsepower for a high-speed engine and probably three pounds per horsepower for a medium-speed engine using domestic fuel. This figure is greatly reduced by proper design, the writer having built air-cooled cylinders of semi-steel for automobile engine use that weighed 1.5 pounds per horsepower. The most efficient aircraft practice does not result in cylinder-body weights of much less than 0.4 pounds per horsepower when operating at a lower speed, but using compression-ratios that are impossible for car practice and fuels that are not at present available for commercial work. Semi-steel is practically never used for cylinder castings in Great Britain though it has been used in this country and in France. A typical analysis of British air-cooled aircraft-cylinder iron is as follows:

	Minimum	Maximum
Combined Carbon, per cent	0.50	0.80
Total Carbon, per cent	2.70	3.50
Silicon, per cent	1.20	2.00
Sulphur, per cent	...	0.12
Phosphorus, per cent	...	0.80
Manganese, per cent	0.50	1.20

An interesting sidelight on cast iron as a cylinder material is seen in the following analysis which is taken from a British report on the materials of construction used in aircraft and aircraft engines. The twenty per cent increase in the conductivity after annealing possibly explains why cast-iron cylinders of very light section rarely perform as well when new as after considerable service.

ANALYSIS OF A CYLINDER IRON

	As Cast	Annealed
Graphitic Carbon, per cent	2.640	3.340
Combined Carbon, per cent	0.850	0.150
Silicon, per cent	1.840	1.940
Sulphur, per cent	0.078	1.090
Phosphorus, per cent	1.090	1.090
Manganese, per cent	0.820	0.850
Conductivity at 212 deg. Fahr., in C. G. S. units	0.102	0.121

Improved Method of Melting Iron.—A new process for melting iron whereby it is freed of gaseous occlusions and consequent porosity, and is also deoxidized, is described in *Der Motorwagen* by Dr.-Eng. Gg. Bergmann. It was observed long ago that when iron is transported over considerable distances while in the molten state, it has a denser structure after solidification. On the basis of this observation the suggestion was made by Dr. Dechesne that the iron should be shaken or jolted while molten. Ex-

periments made to this end showed that even the smallest gas bubbles—which are subject to the high specific pressure of the liquid iron—are detached by the jolting and escape. It appears that certain other recently developed processes for freeing iron of gaseous inclusions or spongy spots call for greatly overheating the melt, to as high a temperature as 2,700-2,900 degrees Fahrenheit, and they also require preheating of the moulds. The overheating naturally involves a large increase in the fuel consumption and the preheating of the moulds is said to be difficult and uncertain. In the new Dechesne process the treatment of the iron is entirely mechanical, and the physical effect is substantially the same as when a bottle of carbonated water is twirled and shaken.

In the course of experiments substantial amounts of steel were added to the melt, and it was found that the resulting iron (semi-steel) did not "smear" or show any hard spots, in which respects it contrasted favorably with other semi-steels. Also the much-dreaded tendency of the iron to turn white was entirely absent. "Air-cooled" cylinders, which when produced by other methods always showed hard spots and a white grain structure in the flanges, were satisfactory in every way and remained gray to the very ends of the flanges. The following table gives the chemical compositions and mechanical properties of "agitated" iron with different additions of steel. The transverse strengths were obtained from unfinished rods $1\frac{3}{16}$ inch in diameter and with 23.6 inches between supports.

PROPERTIES OF SEMI-STEELS

	ANALYSIS									Pour'g. Temp.
Mark	C	Si	Mn	P	S	T.S.	Tr.S.	El.	Bri. H.	Deg F.
R. 0	3.72	2.22	0.46	0.37	0.17	29100	59500	11.5	222	2410
R.10	3.20	2.27	0.56	0.38	0.08	36850	72200	12.5	216	2390
R.15	3.40	2.34	0.45	0.33	0.09	39000	74500	12.5	228	2370
R.20	3.38	2.15	0.42	0.37	0.13	42500	80000	12.5	217	2430
R.20	3.62	1.74	0.71	0.30	0.17	43300	77500	14.2	240	2370
R.30	3.48	2.15	0.45	0.20	0.13	45400	78800	14.0	215	2480
R.30	3.48	1.57	0.68	0.41	0.14	48000	81600	11.9	224	2445
R.50	3.02	1.83	0.40	0.34	0.14	47500	87300	12.5	215	2445
R.70	2.82	2.94	0.66	0.15	0.14	46100	92200	18.0	189	2460

Nichrome Improves Cylinder Iron.—For some time automotive engineers have been seeking some way of producing stronger castings for use in engine cylinders. Because of the irregularity of the castings, the numerous cores required and the necessity for absolutely sound castings, gray iron with a high silicon content has been the best available. Many attempts have been made to alloy this metal in such a way that the strength and hardness would be increased, but considerable difficulty has been experienced in obtaining uniform results. Nickel has been added successfully either in the cupola or in the ladle of molten metal and has made an improvement in the castings. Silicon, however, which is present in relatively large quantities in most automotive castings, co-operates with the nickel in forming large flakes of graphite which produces an extremely soft prod-

uct. To offset the effect of these elements chromium has been added to the mixture. The addition of chromium does break down the granular structure and improve the castings to a considerable degree, but it was found almost impossible to get any uniformity of results in castings made. When chromium is added in the cupola, under the splendid oxidizing conditions present, a considerable portion of the metal oxidizes so that it is always uncertain just what the chromium content of the poured mix will be. When metallic chromium is added to the ladle it is immediately oxidized at the surface of the melt, thus making it impossible, or at least extremely difficult, for the molten iron to attack it. The addition of other alloying substances was tried, but none seemed to offer the same possibilities as

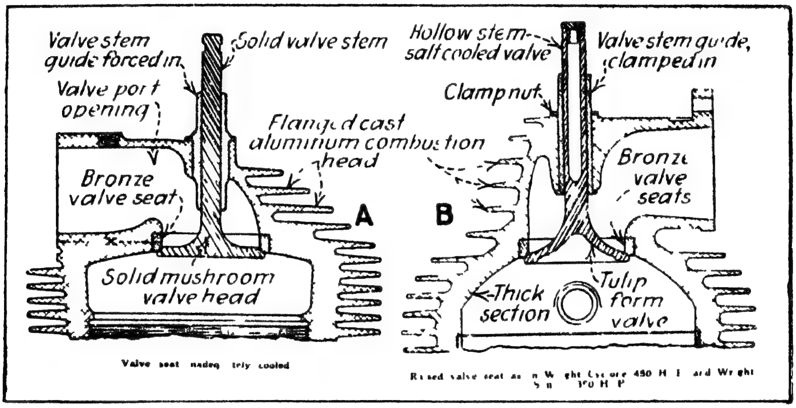


Fig. 294A.—Air-Cooled Cylinder Combustion Head Forms. A—Valve Seat Inadequately Cooled. B—Raised Valve Seat a Superior Form. Note Hollow Stem Valve for Salt-Cooling.

nickel and chromium, so further attempts were made to discover some method by which they could be added to the iron in such a way that the content of the alloys in the castings could be foretold with accuracy.

This search led to the trial of Nichrome. Nichrome has the following approximate analysis:

Nickel 60 per cent
Chromium 12 per cent
Iron 24 per cent

with small quantities of carbon, manganese and other elements. Varying quantities of this alloy were added in the ladle with very satisfactory results. The practice was to add metallic nichrome to a hot empty ladle and then fill it with molten iron from the cupola spout. It was found that the Nichrome melted almost instantly and all its ingredients entered the solution and that the stirring effect of the pouring process effectually mixed the mass so that the alloy was evenly distributed throughout the ladle.

Two per cent of Nichrome by weight added to the iron produced castings which were considerably harder than ordinary gray iron, were more even in texture, had increased tensile and transverse strength, were much more resistant to abrasion, but could be machined with the same feeds and

speeds as were used for ordinary castings. The extent to which the addition of Nichrome influences the hardness of castings is shown in the following table, which gives Brinnell index figures for castings without Nichrome and with varying proportions of that substance:

Amount of Nichrome added	Brinnell hardness
None	207-217
1 per cent	212-228
2 per cent	223-235
3 per cent	241-255

A typical analysis of an iron cylinder casting to which Nichrome has been added is given:

Carbon	3.14 per cent
Manganese	0.55 per cent
Silicon	2.07 per cent
Chromium	0.42 per cent
Nickel	1.57 per cent
Copper	0.10 per cent

QUESTIONS FOR REVIEW

1. Outline requirements of successful air-cooled cylinder design.
2. What determines the temperature of an air-cooled cylinder?
3. How does the temperature vary at different parts of the cylinder?
4. What is the effect of mixture strength on cooling of cylinders?
5. What is the best air-cooled cylinder form?
6. How are cooling fin dimensions determined?
7. Why is circumferential finning best?
8. Name types of air-cooled cylinder heads.
9. What material is used for heads of composite cylinders and how are they fastened?
10. Compare composite and integral cylinder construction.
11. What material is used for air-cooled cylinders?
12. What is the effect of nichrome on cylinder iron?

CHAPTER XXI

AIR-COOLED ENGINE VALVES—VALVE TIMING—CYLINDER FINISHING

Exhaust Valve Cooling in Air-Cooled Cylinders—Consideration of Internal Valve Cooling—Cooling by Way of Valve Seat—Cooling by Way of Valve Stem—Internally Water-Cooled Valve Stem—Fusible Salts Used in Valve Stem—Composition of Salt Filling—Valve Steels—Tulip Form of Valve Head—Roof Head Cylinder—Design of Valve Stem Guides—Valve Seat Inserts—Valve Timing—Advanced Exhaust Valve Opening Essential—Blowing Back—Why Lead is Given Exhaust Valve—Exhaust Closing, Inlet Opening—Closing Inlet Valve—Time of Ignition—No Set Rules for Valve Timing—How an Engine is Timed—Timing Gnome Rotary Engines—Finishing Cylinder Bores—Best Speed of Rotation for Grinder Head—Advantages of Honing and Lapping Cylinder Bores.

Exhaust-Valve Cooling in Air-Cooled Cylinders.—In general, exhaust-valve cooling has two distinct phases, cooling through the seat and cooling via the stem and the guide. If either be at fault, inspection when running on open exhaust will determine which is the offender. For efficient seat-cooling as effected by valve design, most aviation engine designers prefer a tulip valve with a thick rim, possessing greater circumferential conductivity than the flat-head valve, and the use of a wide valve-seat. The width of the valve-seat undoubtedly has a marked effect upon the heat-flow from the valve to the cylinder-head, as the intensity of the heat-flow will depend on the area of the valve in contact with the cylinder. The heat-flow from the valve to the cylinder will be affected to some extent by the load of the valve-spring; from this standpoint the greater the load the better. The Type J cylinder shown at Fig. 294 exhibits very good seat cooling, and uses a wide valve-seat, the bore of the exhaust port being 2¼ inches and the top diameter of the seat 2⁹⁄₁₆ inches. Valve-seat cooling in this cylinder according to Mr. Heron is as good as obtains in any water-cooled cylinder using a similar valve size and under all running conditions a dead black rim ½ inch wide is shown on the valve.

Efficient stem-cooling is assisted by extending the valve-guide and guide-boss down as closely to the head of the valve as possible, and shrouding the guide with a heavy boss that is able to conduct away the heat abstracted from both the valve-stem and the exhaust gas. The cooling viâ the valve-stem is partly controlled by the sectional area of the stem and the area of the stem in contact with the guide.

The Type J cylinder is of efficient design as regards conducting the heat away from the exhaust-valve guide. This cylinder initially had a ⁷⁄₁₆-inch diameter exhaust valve-stem and this was found to have insufficient area to carry away the heat from the head of the valve and insufficient surface to dissipate the heat from the stem to the guide efficiently. As a result the stem overheated; further, the high-temperature zone extended too far up the stem. An increase in the stem diameter has improved the cooling to some extent. Partially as a result of the stem overheating, trouble was experienced with breakage. Claims that the exhaust-valve

cooling of air-cooled engines is equally as good as that of water-cooled engines may appear to be an exaggeration. Nevertheless such claims are not merely the personal opinion of Mr. Heron but can be confirmed readily by the Engineering Division of the Air Corps.

With good valve-cooling it is possible to maintain a black rim on the valve-head under all conditions. With 1½-inch diameter valves the maximum temperature of any part of the valve should not exceed 1,200 degrees Fahrenheit, and with valves up to 2½ inches in diameter it is possible to avoid temperatures in excess of 1,350 degrees Fahrenheit, although this is at times accomplished at the expense of fuel economy. In general, for air-cooled engines Mr. Heron prefers to avoid the use of exhaust valves in excess of two-inch diameter. Internal valve-stem cooling by water or mercury has been used with varying degrees of success. However, in the opinion of some authorities, equally efficient results generally can be obtained with less expense and complication by greater care in the cooling design of the cylinder body. The amount of oil reaching the combustion-chamber has a considerable influence on the valve temperature, and an increase in the oil supply is often sufficient to reduce the exhaust-valve temperature to the point at which reliability is obtained.

There are several probable reasons given by Mr. Heron why an air-cooled cylinder should have as good exhaust-valve cooling as a water-cooled cylinder. These are (*a*) the metal sections of the head and the valve ports have sufficient heat-flow capacity to carry heat to the cooler portions of the head; (*b*) the intensity of the heat-flow to the cooling medium is lower on account of the area of the attached fins; (*c*) with good design, the cooling medium surrounding the exhaust-ports and the adjacent portions of the head is in a violent state of turbulence, and semi-stagnation of the cooling medium, which may occur in a water-cooled engine, cannot well occur in an efficient air-cooled cylinder; (*d*) no equivalent of a steam-pocket can exist and local overheating does not result in practically complete failure of cooling at that point, such as occurs when a steam-pocket develops in a water-cooled cylinder; (*e*) no equivalent of lime-salt deposits and resultant heat-insulation can occur; and (*f*) a parallel to an insulating film of steam does not exist. If a valve in an air-cooled cylinder runs cooler than a valve of equal size in a similar water-cooled cylinder, it is not unreasonable to assume that the exhaust port and the seat are cooler in the former. From observation it has been found that it is not possible to maintain a wetted surface with semi-stagnant water on metal at anything like the temperatures, 500 degrees Fahrenheit, developed around the exhaust-seat of an air-cooled cylinder. Water will remain almost quiescent and in apparently close contact with metal at 500 degrees Fahrenheit for a considerable time with little evident evaporation or transfer of heat from the metal to the fluid; a 50 per cent reduction in the temperature of the metal, however, will produce violent evaporation. Enclosing the valve-stems and the guides and lubricating them with oil improves exhaust-valve cooling. This alone is sufficient ground for a positively lubricated valve-gear, apart from other obvious advantages.

In its original form, the type-J cylinder developed 48 brake horsepower at 1,650 r.p.m. Continual trouble with exhaust valves and guides

was experienced. The exhaust valve-stem at the junction of the stem and the neck was bright red, about 1,300 degrees Fahrenheit, and invariably fractured there after a relatively short period of running, tungsten-steel valves lasting about 25 hours and stainless-steel valves about six hours. Cast-iron exhaust-valve guides wore out in less than ten hours. A file-hard tungsten-steel valve of the type shown in Fig. 295 A having a 9/16-

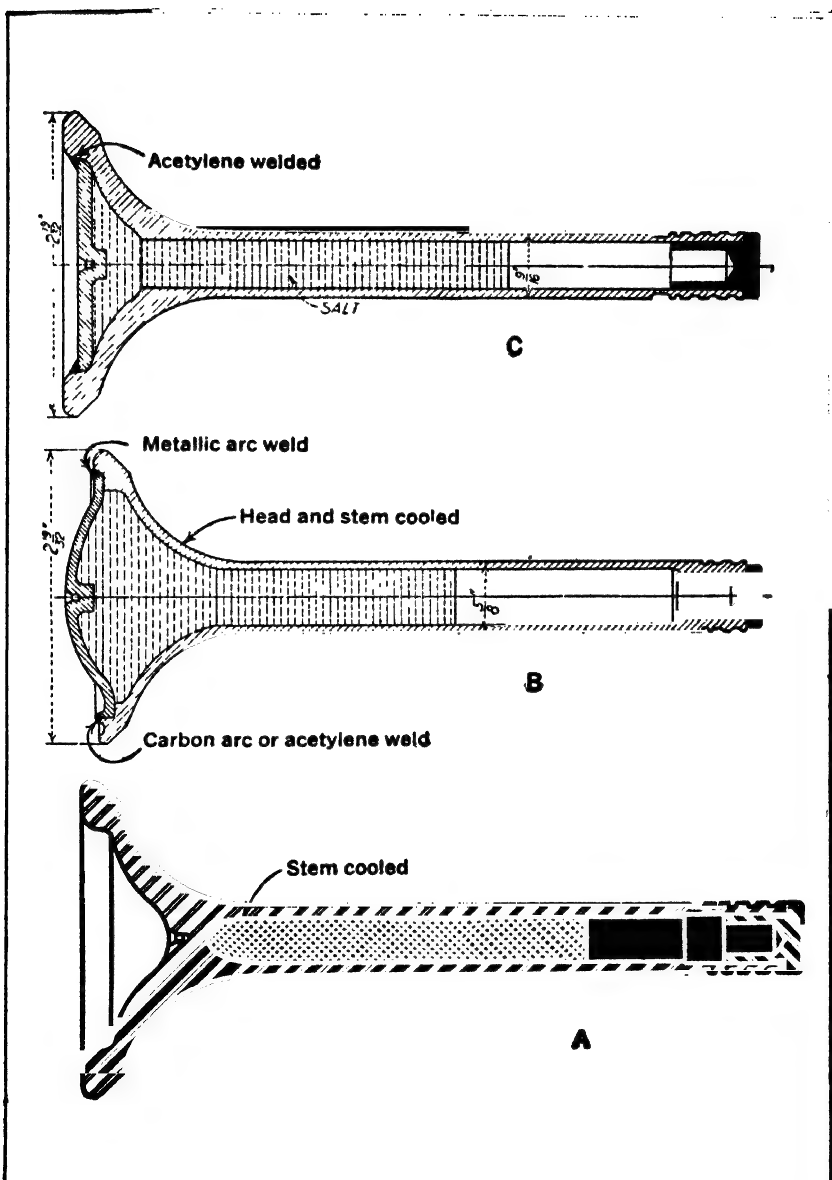


Fig. 295.—Internally Cooled Exhaust Valve Stem Shown at A. Internally Cooled Valve Heads and Stem Shown at B and C.

inch stem, but with a shallower hole and no filling, was then tried and eliminated breakage; at least, in a 50-hour full-throttle test. It scaled, however, on the stem and the neck as the result of a temperature of approximately 1,250 degrees Fahrenheit. In this test, a hard No. 3,440 S. A. E. Steel valve-guide was used, proving much superior to cast iron, but wearing considerably at the valve-tip end on account of side-thrust and slightly at the head end because of the scale on the valve-stem.

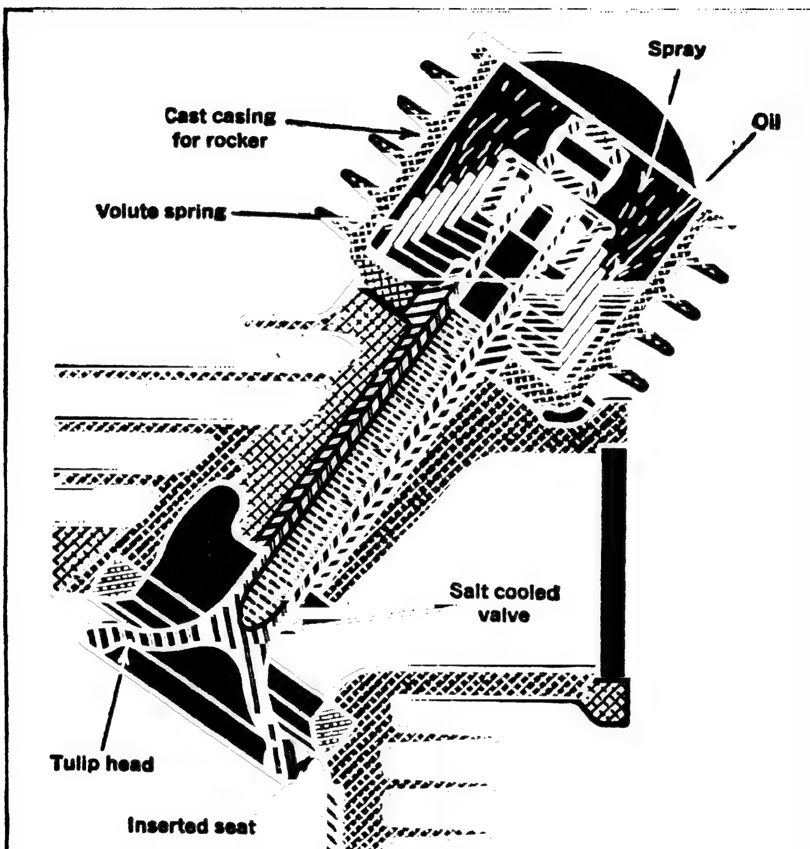


Fig. 296.—Enlarged Section Through the Exhaust Port of Type K Aircraft Engine Cylinder Showing Housings for Valve Spring and Oil for Cooling the Spring and Insuring Lubrication of Valve Stem Guide.

A file-hard tungsten-steel valve of exactly the type shown in Fig. 295 A was then tested with salt, using a hard tungsten-steel guide. The valve-cooling was much improved. The stem was dead-black and the hot zone, although reaching a temperature of approximately 1,100 degrees Fahrenheit, was much reduced in area, being about ¾ inch wide and confined to the neck midway between the inner edge of the seat and the junction of the stem with the neck. After 100 hours of full-throttle running, the last 50 hours being under conditions of violent auto-ignition without attention, scaling of the neck was evident, but very little was observed on the stem.

Wear of the valve-stem and of the hard tungsten-steel guide was found. This was definitely due to the solid tappet, as later tests in which the rocker fulcrum was altered to a more favorable position relative to the valve-tip did not greatly reduce the wear of the guide, valve-stem, tip and tappet. Substitution of a roller tappet, as shown in Fig. 294, finally eliminated the wear of all these parts. Consequently, they will now stand a 100-hour full-throttle test at 1,800 r.p.m. without measurable wear. Tests on this cylinder, when fitted with a case-hardened inlet-valve, showed that extreme hardness is of advantage even for inlet-valves.

Much work has been done by Mr. Heron on the Type-K air-cooled cylinder. This cylinder is of 4½-inch bore by 5½-inch stroke and has a 1⅞-inch diameter tulip inlet-valve and a 1$^{13}/_{16}$-inch diameter tulip exhaust valve, both with ½ inch lift. A section of the exhaust port of the type-K cylinder is given at Fig. 296. Tests were carried out with a cylinder that had a cast-iron head and in which the bronze valve-seat inserts and spark-plug bushings were omitted. This cylinder developed 140 pounds per square inch brake mean effective pressure or 27.8 brake horsepower at 1,800 r.p.m., the temperature of the head wall at a point on the exhaust-valve seat reaching 750 degrees Fahrenheit and even higher. Despite this very high head-temperature, excellent cooling was obtained with a salt-cooled valve, the hottest zone on the valve being in the middle of the neck, about ⅜ inch wide, and just perceptibly red, approximately 900 degrees Fahrenheit. With an unfilled valve identical in design with the one shown in Fig. 296 the valve-cooling was by no means so good; the red zone began about ⅜ inch from the inner edge of the seat and extended over all the visible portion of the stem, the maximum temperature of 1,300 to 1,350 degrees Fahrenheit occurring at the junction of the stem and the neck.

While so high a temperature is undesirable, the design was partly responsible; improved results could no doubt have been obtained in this case with a valve designed to be used without internal cooling. Notwithstanding, the cooling was better than that obtaining in some water-cooled aircraft-engines now in use. These tests indicate that cylinder temperature has far less influence on valve-cooling than is commonly supposed and that the major factors in valve-cooling are the design of the valve, valve-port, valve-seat, guide and guide-boss. It is notable in these tests that, in spite of the high temperature of the exhaust-valve seat in the cylinder, the exhaust valve rim was black, less than 900 degrees Fahrenheit, in all tests, showing that a high rate of heat-transfer from valve to cylinder is possible where the temperature difference of the portions in contact is less than 150 degrees Fahrenheit. At the conclusion of the above-mentioned tests, all fins were removed from the head and the barrel of the cylinder, and a water-jacket was welded into place. No difference in performance, 27.8 brake horsepower or 140 pounds per square inch brake mean effective pressure at 1,800 r.p.m., was obtained, but the valve-cooling was improved. A salt-cooled valve ran dead-black under all conditions, while with the uncooled valve previously used, a red zone about ⅜ inch wide having a temperature about 1,100 degrees Fahrenheit was evident at the junction of the stem and the neck.

Consideration of Internal Valve-Cooling.—The main purpose of internal valve-cooling is to reduce the temperature of the valve-head and of the head end of the valve-stem. This is done by reducing the temperature differences within the valve, or, in other words, by reducing the temperature of the head and increasing that of the tip end of the stem. In the stem a reduction of the maximum temperature and of the temperature gradient from the head end to the tip end is of advantage, as an increase of the mechanical strength and of the fatigue resistance of the head end results. Elimination of red-hot surfaces on the stem produces better bearing conditions within the guide. An increase of temperature at the tip end of the valve makes available a greater temperature-gradient for transmitting heat to the outer end of the guide, to the oil, if used, and to the cooling medium. Elimination of the red-hot head of the valve should be sought, as the relatively large incandescent area of a normal exhaust-valve tends to produce detonation and thus to limit the available compression-ratio.

Usually, the result of any extensive red-hot area in the head of an exhaust-valve is to render the valve generally unreliable. Should the red zone extend to the rim of the valve, pitting and burning of its seat usually develop very rapidly in service. Deterioration of the valve itself under the latter condition is often accompanied by the cutting, hammering-down and pitting of the seat in the cylinder-head. High temperature in the valve neck does not necessarily involve danger of damage to the valve-seat. In some air-cooled cylinders, despite fairly high temperatures in the valve neck, pitting or burning of the valve or the cylinder seats does not occur, the rim of the valve remaining black under all conditions. It may be well to state that in the latter case the cooling of the valve rim is in no way due to the cylinder's being air-cooled, but rather to the design of the valve rim and seat; in fact, with efficiently applied water-cooling, the conditions at the rim and the seat are somewhat better.

Cooling by Way of Valve-Seat.—Valve-cooling has two principal phases: cooling by way of the seat and cooling by way of the stem and the guide. The width of a valve-seat has a marked effect upon the efficiency of the seat-cooling obtained. A wide seat enlarges the area available for heat-flow, thus both increasing the amount of heat flowing by this path from the valve to the cylinder and reducing the temperature-difference between the rim of the valve and the cylinder, on account of the lower intensity of the heat-flow. Cooling by means of the stem and the guide is benefited by several design factors, namely: large diameter of the valve-stem, bringing the guide as close to the head of the valves as possible and shrouding the guide with a heavy guide-boss. Fig. 296 which shows a section through the exhaust valve, port and guide of the Type-K cylinder, illustrates the most successful designing practice of the Engineering Division with regard to exhaust-valve cooling. It embodies all the design factors mentioned above. The large-diameter valve-stem is of advantage in that it increases the area available for heat-flow from the stem. The possibilities of heat dissipation through the stem are evident if the ratio between the contact area of the valve-seat and the cylinder and the contact area of the stem and the guide are considered. In the Type-K cylinder, the contact area of the seat is 1.075 square inch and that of the stem with the

guide is 4.570 square inches, a ratio of 4.25 to 1; and this takes no account of the further stem-cooling that is due to the tip's being sprayed with oil. In the Type-J cylinder, the seat area is 1.67 square inch and the stem area is 5.47 square inches, a ratio of 3.27 to 1. These two ratios show that, if suitable means be used to conduct heat up the stem and maintain a more or less uniform temperature-distribution throughout its length, much more heat can be dissipated by the stem than by the seat.

Cooling by Way of Valve-Stem.—While the contact between the stem and the guide is less intimate than that between the valve-seat and the cylinder, the latter contact is maintained for only two-thirds of the cycle, approximately; during the remaining third, the valve-seat and the cylinder are not only out of contact, but the valve rim is rapidly receiving heat from the outgoing gas. The effectiveness of thermal contact between the stem and the guide is markedly increased by the oil-film resulting from lubrication, for, although it is a poor conductor, oil produces a much better thermal condition than that produced by dry metal-to-metal contact with its intervening air-gap between the surfaces. In the preceding calculations of stem contact-surface, the area of the whole circumference is given. It is theoretically possible to obtain this amount of contact with running fits but, nevertheless, with the small clearance of the stem in the guide rendered possible by the salt-cooled valve and the hard valve-guide, this condition substantially obtains even without lubrication. Some present designs using lubricated valve-stems would function much less satisfactorily with dry valve-stems and guides. Internal cooling renders available for heat dissipation the relatively large surface of the valve-stem in a way that no possible solid stem can. A body of liquid in a violent state of turbulence contained within the interior of the valve conducts heat much more rapidly by virtue of its motion than can a solid metal stem having a cross-section equal to that of the column of liquid.

Internal Water-Cooled Valve-Stem.—Some previously used methods of internal valve-cooling have relied upon the direct dissipation of heat from the valve-stem to the outside air by a finned valve-stem beyond the guide. A design of this type is shown in Fig. 297 which illustrates a water-cooled valve applied to an R. A. E. 4E air-cooled cylinder. The experience of the Engineering Division has shown that so elaborate a design is entirely unnecessary for efficient internal-cooling. Internal-cooling is not considered worth the complications of split valve-guides, forked valve-rockers and the increased headroom necessitated by added valve-length, all of which are introduced in a design of the type shown in Fig. 297. Such complications result in unsatisfactory design with regard to the rocker side-thrust on the valves, practically eliminating any possibility of completely enclosing the valve-guide, spring, rocker and push rod. Efficient internal valve-cooling sharply reduces the maximum valve temperature. And there is good reason to believe that a considerable reduction in the mean temperature is obtained, although this is not of marked importance, the main consideration being reduction of the maximum temperature.

Fusible Salts Used in Valve-Stems.—In the Engineering Division's salt-cooled valve, cooling is obtained by partly filling a hollow valve with

a mixture of fusible salts that has a boiling or decomposing point well above any temperature attained by the valve when it is in contact with the mixture. This method avoids the difficulty of retaining the cooling fluid within the valve that is experienced in this county with mercury-cooled valves and in England with water-cooled valves. With salt fillings, the pressures attained in the interior are small and are readily contained by a drive-fit plug in the tip of the stem. The final seal by welding, required with valves in which the cooling is produced by the boiling and the condensing of a fluid, is in this case unnecessary.

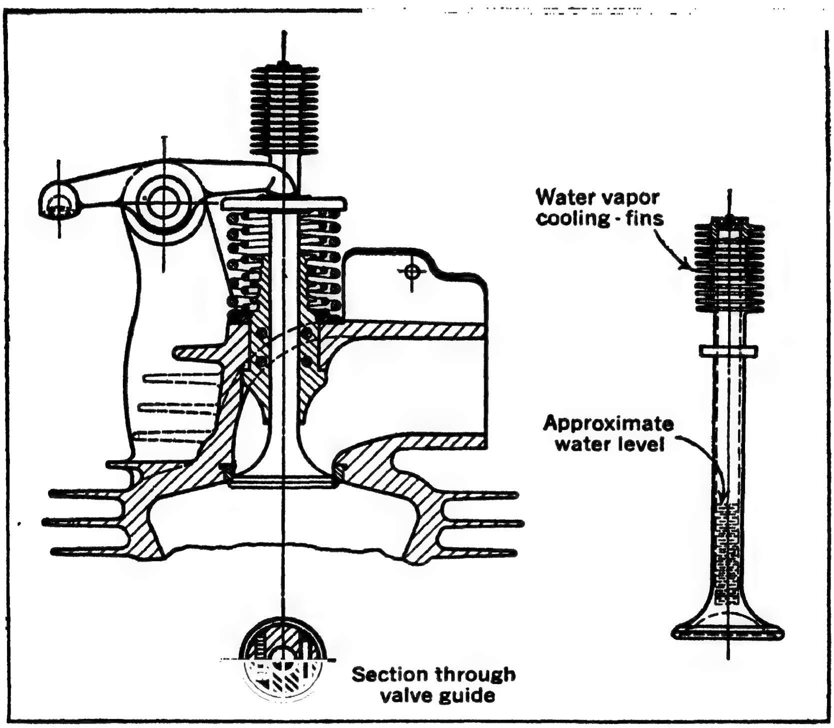


Fig. 297.—Application of a Water-Cooled Exhaust Valve to an Experimental Air-Cooled Engine Cylinder. This Method of Cooling Requires Very Elaborate Design.

With mercury or with water-cooled valves, cooling is obtained by the enclosing in the stem of a small quantity of fluid, which is boiled at the head end and condensed at the tip end. To work most efficiently, this type of valve should have a vertical stem and its head should be down. Otherwise, if the stem be far out of the vertical, the boiler and the condenser action will fail. Although the cooling of valves filled with fluids vaporizing at working temperature is due mainly to boiling and condensing, under certain conditions a considerable amount of cooling seems to be obtained by the turbulent motion of the filling. There is reason to believe that the equality of valve-cooling between the top and the bottom cylinders experienced with mercury-cooled exhaust valves in some air-cooled radial engines is as

much due to the turbulent motion of the mercury as it is to the boiling and the condensing, for about 60 per cent of the stem volume is filled with mercury, which is in excess of the amount required for boiling and condensing only. The valves in the lower cylinders obviously would fail to cool by the latter method if they were filled with only sufficient fluid for this purpose, as the fluid would be in the condenser and the vapor in the boiler.

The internal pressure produced by the boiling of mercury or of water causes much trouble from leakage. The mercury-cooled valve has never been dependable, for it is by no means unusual to find after some hours of service that a considerable portion of the mercury has leaked out of the stem. With water-cooled valves, great difficulty has been experienced with leakage, and also with the high internal-pressure's either swelling or bursting the valve-stem. Pure tin, the first filling tested by the Engineering Division, gave satisfactory cooling but so eroded the interior of the stem at the head end that a fracture resulted.

It is not surprising that troubles with bursting are experienced with water-cooled valves as temperatures in excess of 700 degrees Fahrenheit have been measured on both the head and the tip of such valves. At this temperature, saturated steam develops a pressure of approximately 3,000 pounds per square inch. As such a pressure sets up considerable additional stress in the metal of the valve, the difficulty of containing it by welding or other means is apparent. The Engineering Division's salt-cooled valve does not depend for its cooling upon boiling and condensing, but rather, as stated previously, upon the heat-transfer resulting from the turbulent motion of a nonboiling liquid contained within the interior. The valve is so filled with salt that a space of from 5/8 to three inches or more is left between the top of the molten salt, at the assumed working temperature of 750 degrees Fahrenheit, and the inside end of the valve-stem tip-plug.

Composition of Salt Filling.—The filling that has been most used is the eutectic mixture of sodium and potassium nitrates, which has 45.5 per cent of sodium nitrate and 54.5 per cent of potassium nitrate by weight. This mixture melts at 425 degrees Fahrenheit and, from the melting-point to 750 degrees Fahrenheit, expands approximately sixteen per cent. It readily wets the surface of hot steel, up to 900 degrees Fahrenheit at any rate, and has shown no tendency to decompose in the interior of valves subject to very heavy duty. This filling gave excellent results, but, after examination of the temper colors on the tip end of the valve, it was thought that the melting temperature was too high and that the salt was frozen in the tip of the stem, thus markedly reducing the heat dissipated from this portion of the valve. To overcome this supposedly frozen condition of the salt in the tip of the stem, a filling with a lower melting-point was investigated. The eutectic mixture of 34 per cent of lithium nitrate and 66 per cent of potassium nitrate by weight melts at 265 degrees Fahrenheit but is very hygroscopic, making it necessary to heat the mixture to at least 750 degrees Fahrenheit to get rid of the water.

For an output of 52.5 brake horsepower the valve shown in Fig. 295 A, proved to have insufficient cooling to prevent scaling. Red scale was produced on the neck, particles of which became detached and adhered to the

seat of the valve. Although the stem of the valve was dead-black, a layer of black oxide formed at the junction of the stem and the neck and resulted in poor bearing conditions at the mouth of the head end of the guide. Tests of valves with direct head-cooling were therefore carried out, these valves being of the types shown at Fig. 295 B and C. It is possible that in air-cooled cylinders of large output, head cooling as well as stem cooling may be necessary.

If hollow-head valves are to be used, it will be necessary to secure absolute reliability in any welds that are exposed to explosion pressure. Otherwise the design with stem-cooling only is preferable, there being practically no risk from leakage of the filling with this type.

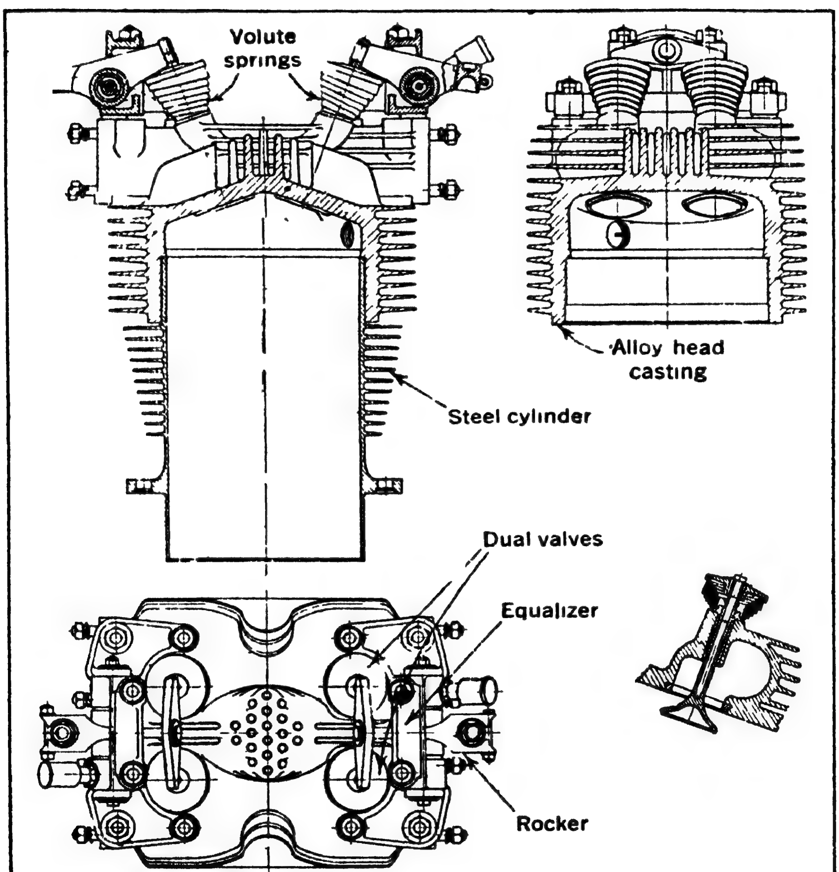


Fig. 298.—Plan View and Sectional Elevation of the Airco Five and One-Half by Six Inch Roof Head Cylinder, Having Four Valves.

Valve Steels.—The Engineering Division up to the present time has found that, for internal cooling, high-tungsten steel, containing from fourteen to eighteen per cent of tungsten, has the advantage in comparison with other materials. With internal cooling, it is possible to eliminate

scaling and burning, which are the chief disadvantages of tungsten steel for exhaust-valves. The advantages of high-tungsten steel are the extreme hardness that is retained during and after exposure to the highest working-temperatures attained with salt-cooled valves, strength at high temperatures and excellent resistance to wear. Strength at high temperature has little advantage with internal cooling; thus, hardness and resistance to wear are the two important considerations. Practically file-hard valve-stems that will not soften in service allow the use of file-hard valve guides, the advantages of which will be discussed later.

A valve steel that could be produced with nonscaling properties in conjunction with secondary hardness or, at any rate, practically file-hardness, after tempering at 750 degrees Fahrenheit, the filling and approximate running temperature, would be superior to tungsten steel for internally cooled exhaust-valves in cases where the maximum temperature of the valve exceeds 900 degrees Fahrenheit. The Engineering Division is investigating the problem of procuring such a steel. The best method of attacking the exhaust-valve problem, however, appears to be by reducing the temperature to such a point that scaling will not occur with any steel. The excellence of modern valve-steels has considerably obscured the exhaust-valve problem, instead of eliminating the cause of the trouble, that is, excessive temperature. A temporary cure has been obtained by improving the properties of the valve material at high temperature. Such an expedient in no way eliminates the fundamental cause, and recurring trouble is experienced with every increase of valve size and duty.

Tulip Form of Valve Head.—The Engineering Division has reason to consider, in general, that the tulip type of valve is superior to the flat-head or mushroom type as regards cooling, gas-passing capacity and resistance to warping and stretching. These conclusions have, as a whole, been corroborated by the Bureau of Aeronautics, of the Navy Department, after much comparative testing of Liberty engines, and by the Wright Aeronautical Corporation, which has carried out extensive tests on Wright E and H engines. Recent full-throttle tests of Liberty engines at the Engineering Division have sharply demonstrated that the tulip form of valve has greater reliability and strength than has the mushroom type.

Valve Size Considerations.—For air-cooled cylinders, the Engineering Division successfully uses valves of relatively smaller diameter, but with considerably higher lift, than are current practice in this country. A comparison of Liberty and Type-J Engineering Division cylinders shows the different types of practice. The Liberty is of five-inch bore and seven-inch stroke, has a capacity of 137.5 cubic inches, runs normally at 1,700 r.p.m., and has two 2½-inch valves. The exhaust valve has a ⅜-inch and the intake valve a 7⁄16-inch lift. The Type-J cylinder is of 5⅝-inch bore and 6½-inch stroke, has a capacity of 161.5 cubic inches, runs at a normal speed of 1,800 r.p.m., and has a 2½-inch intake valve and a 2¼-inch exhaust valve, both with 9⁄16-inch lift. The mean effective pressure of the Type-J cylinder is not inferior to that of the Liberty engine when both are tested as single cylinders; in fact, few cylinders with appreciably higher mean effective pressure than that of the Type-J engine when so tested are known. The practice of using exhaust valves of a diameter

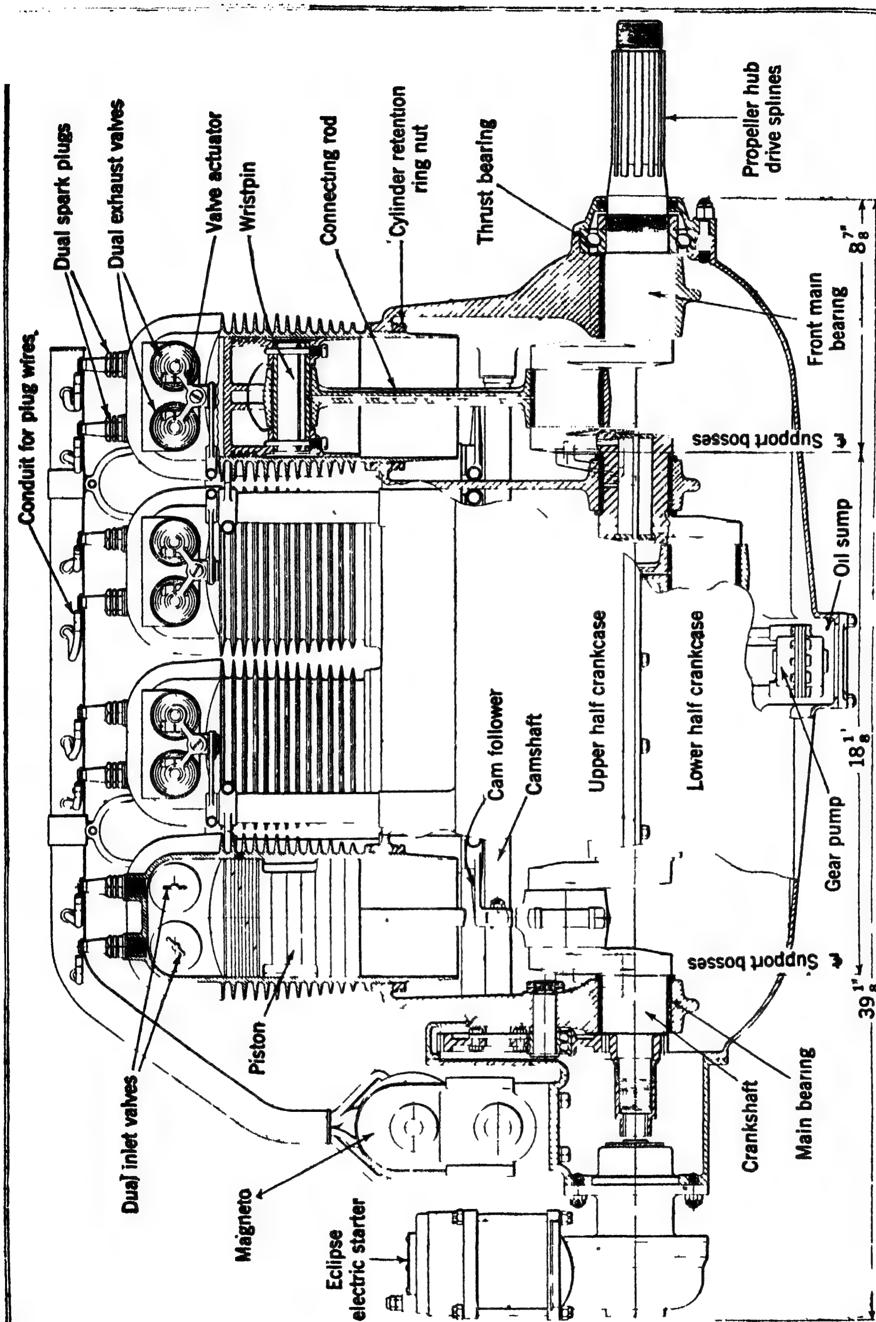


Fig. 299A.—Longitudinal Sectional View of Cameron 60 Horsepower Four-Cylinder Aviation Engine, Showing Dual Inlet and Exhaust Valves and Method of Actuation by Vertical Rocker Shafts.

smaller than that of the intake valves has much in its favor and appears to be free from undesirable effects on performance, as in large two-valve cylinders the exhaust valve is often made smaller than the inlet for valve-cooling reasons with satisfactory results. Such an arrangement usually requires an excessive exhaust-opening period to produce the best results, however, in spite of running generally at a somewhat lower speed than the four-valve type.

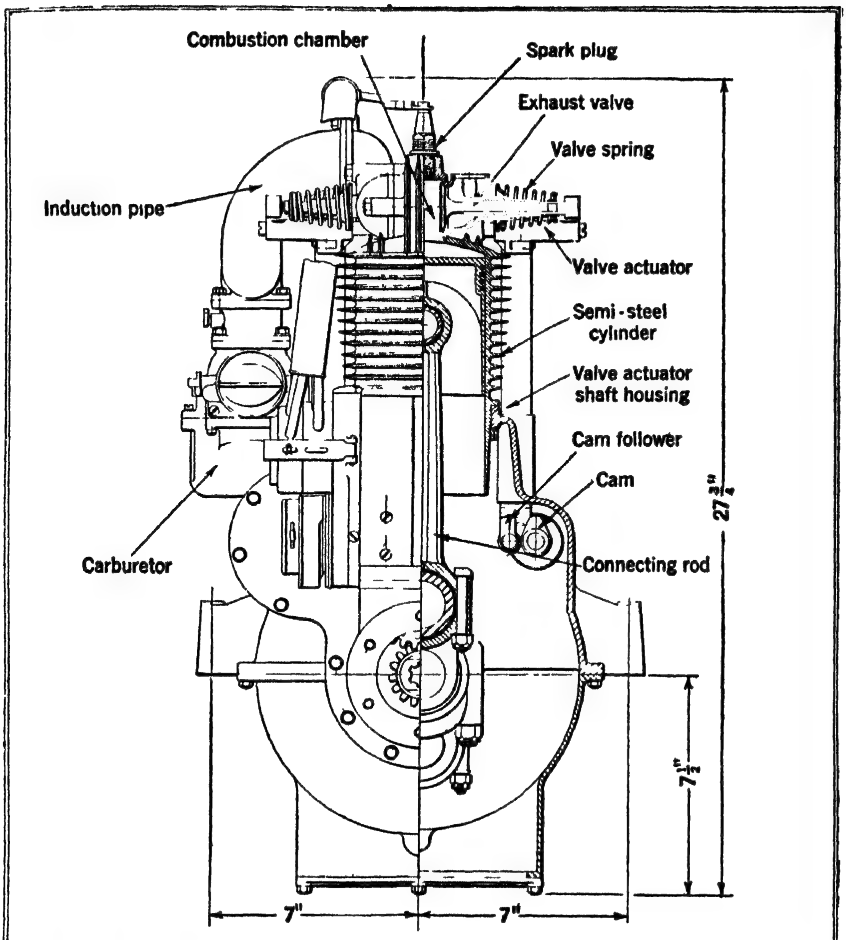


Fig. 299B.—Part Sectional End View of Cameron Air-Cooled Engine Showing Unconventional Type of Cylinder Construction and Placing of Valves so Exhaust Valves are Cooled by the Entering Fresh Gas.

Experience tends to show that with an increase of the valve size the velocity through the annulus can be increased considerably without affecting the brake mean effective pressure. This is probably due to the decrease in friction in the port, owing to the increase of the ratio between the port-area and the maximum area in the annulus, as is brought out in some later remarks on the Type-J cylinder. A large two-valve cylinder, having

gas velocities approximating those of the four-valve type used in the tests, is very difficult to produce successfully. Large valves, if crowded into the combustion-chamber, will result in cooling troubles, and a decrease of the gas velocity by an increase of the lift beyond ⅝ inch generally results in severe mechanical difficulties. In the three- and four-valve types of cylinder, such valving as is used by some American water-cooled aircraft-engine constructors leads to serious cooling difficulties, the principal source being the bridge between the exhaust-valves. The development in Europe of the air-cooled cylinder of high output has resulted in the adoption of relatively small valves used in conjunction with high lifts. Attempts to valve air-cooled aircraft-engine cylinders, particularly of the three- and four-valve types, on the proportions used for air-cooled racing motorcycle engines have resulted in failure; it is by no means uncommon to find the latter type of cylinder in the 30-cubic inch size fitted with four overhead valves of approximately 1½-inch diameter in the port. Such design is possible only with small cylinders and with engines that do not run on full throttle for long periods. Experience tends to show that even for intermittent short bursts at full throttle, the engine having high gas velocities and good valve-cooling will give better power and performance than the type having a combustion-chamber consisting mainly of valves, and that, in spite of the marked advantages of overhead valves for air-cooled engines, their advantage is mainly lost if over-valving is attempted.

Roof Head Cylinder.—The Airco cylinder shown at Fig. 298 is what is known as a roof head four-valve type and tests have demonstrated that its power output was not greatly superior to the two-valve cylinder of the same size. The use of two exhaust valves instead of one usually results in superior valve cooling. In the Cameron four-cylinder engine design shown at Fig. 299 A cooling of the exhaust valves, which are of the solid stem type is obtained by their location in the cylinder head where they are directly opposite the intake valves and all fresh gas impinges directly on the exhaust valves, this producing two beneficial results in that the fuel is completely vaporized and heat abstracted by any liquid globules in the vapor is taken from the exhaust valve heads and reduces their temperature The valve stems are horizontal with this cylinder head design and valves are actuated by a patented vertical rocker shaft arrangement. No compensation gear is needed as the lengthening of the cylinders due to heating does not affect the clearance between the valve stems and operating means, as is the case with overhead valves having inclined or vertical valve stems. The end sectional view at Fig. 299 B shows the type of combustion-chamber employed that makes opposite valve placing possible in Cameron engines. The head construction promotes turbulence in fresh gas during compression stroke.

Design of Valve-Stem Guides.—The problems of wear and scoring of the valve stem and the valve guide seem to be worthy of much investigation. Examination of the valves and the guides from service engines after normal use has shown that, even when lubricated, normally ten per cent of the stems and the guides are scored by the time the first overhaul becomes necessary. The obvious line of attack is to increase the hardness of the wearing surfaces; and this has in practice proved to be one successful

method of solving the difficulty. File-hard tungsten-steel salt-cooled valves are now on test with both file-hard tungsten-steel and case-hardened low-carbon-steel guides. The tungsten-steel guide is probably the best general solution of the valve-guide problem; for, although it is not quite so hard initially as the case-hardened guide, it is less likely to lose its hardness in service on account of cylinder temperature, removal or insertion by a blow-pipe, in aluminum cylinders, or any heat-treatment that is likely to be given to the cylinder-head during the assembling and the enamelling.

The case-hardened guide gives excellent service where conditions do not cause loss of hardness at the valve-tip end. But when practically the whole of the guide is contained within the guide-boss of an air-cooled cylinder, the entire guide becomes partly softened. In water-cooled cylinders, and in air-cooled cylinders, when a considerable portion of the top end of the guide protrudes from the boss and is thus directly air-cooled, or when the guide is lubricated and thus oil-cooled, the loss of hardness at the valve-tip end of the guide is inappreciable. Drawing of the valve-head end of the guide occurs more or less in any case, but in practice it is of no consequence, as little or no wear occurs at this point, and, unless the valve-stem scales on account of excessive temperature, galling of the hard valve-stem does not occur. Extreme hardness of the valve-tip end of the guide is required to resist the side-thrust occurring with rocker-operated valves. Wear of the valve-stem and of the guide due to side-thrust and the resultant tilt of the stem in the guide first occur on a line at the mouth of the guide and over a length of the stem. Line contact of the stem at the mouth of the guide ceases as soon as wear takes place. It is largely this line-contact that results in the cutting and the galling of soft valve-stems; the greater the clearance between the stem and the guide and the shorter the guide, in terms of the number of valve-stem diameter lengths, the greater will be the tendency to tilt and to cause cutting and galling.

The arguments against the tungsten-steel guide are the cost of the material and the difficulty of machining. To ream a hole so free from scratches and grooves that lapping-out 0.001 inch after hardening will produce a mirror-like finish is difficult. If the surface in the bore before hardening is such as to preclude lapping for a final finish, grinding must be resorted to. This is both difficult and expensive with the small bores used in valve-guides. If reaming and lapping are depended upon, a badly reamed hole will result in a waste of expensive material. It is possible that the heat-treatment of the bar-stock might reduce the difficulty of reaming a hole free from scratches. The possibilities of rifle-drilling followed by broaching, as a commercial method of producing a smooth finish requiring only lapping after hardening, are being investigated.

The case-hardened guide produced from No. 1015 or No. 1020 S.A.E. Steel has many manufacturing advantages. If a poorly machined hole is produced, the cost of the scrapped material is negligible. After heat-treatment, it is necessary to remove only 0.001 inch by lapping to produce a mirror-like finish in the bore. The Engineering Division's practice is to ream 0.0005 inches, nominal, under standard size, to harden and then to lap to 0.0005 inches, nominal, over standard size. When the guide is driven

into the cylinder, the bore returns to practically standard size. Case-hardened guides are given a case-hardening $\frac{3}{64}$ to $\frac{1}{16}$ inch deep, a practice that is in accord with that of the Engineering Division in connection with wearing parts subject to heavy loads in air-cooled cylinders.

The Randall graphite-filled valve stem guide shown at Fig. 300 is a real step forward in the perfection of the poppet valve internal-combustion motor. Hitherto no satisfactory manner of lubricating the valve stems has been devised. Oil cannot always be used for lubrication as the exhaust valves may become too hot and the oil will carbonize on the stems thereby

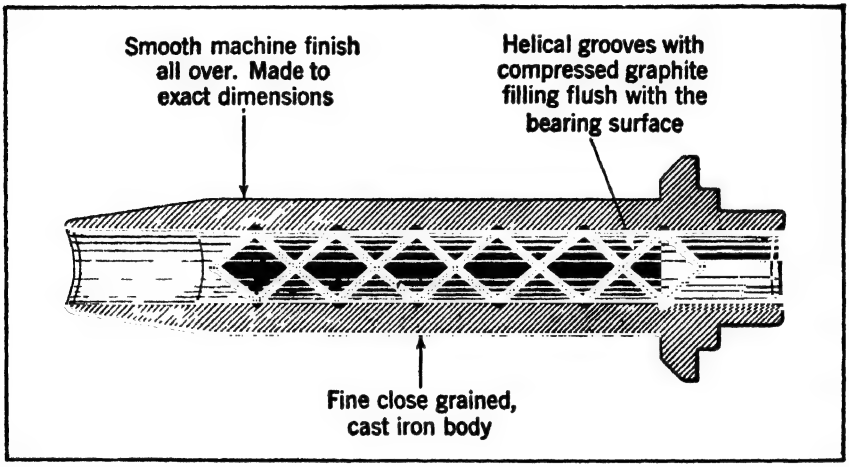


Fig. 300.—Sectional View of Randall Graphite-Filled Grooved Valve Stem Guide.

increasing friction and causing sticking valves. This condition makes it desirable to use a solid lubricant which is not affected by the great heat conducted through stems to guides. The only substance which efficiently fulfills all requirements is said to be a graphite mixture. Using graphite in various forms as a lubricant is very old. Millions of oil-less bearings made under the same patents as these guides are giving service throughout the automotive industries so satisfactorily that their uses and applications are constantly increasing. The application of graphite to the valve-stem guide with this patented process lubricates this part and practically prevents friction. To eliminate friction means longer wear, less power consumed, more power delivered, and what is even more important the whole unit, guide and valve remaining in line will assure the valve head being always tightly seated. It is claimed that a valve stem working in the graphited guide will keep the valve seated as well after thousands of miles of use as it does in a new motor and these guides have been widely used as a replacement in motors of the OX series as well as others of the water-cooled form where no provision is made for adequate valve-stem cooling or lubrication.

Valve Seat Inserts.—When aluminum alloy is used for cylinder heads, it is evident that this material would not permit of seating valves directly

in it because it is too soft and the valve seatings would soon pound out. The proper application of valve seat inserts of harder material is somewhat of a problem. Inserts have been cast in, shrunk in, threaded in and expanded in, the latter construction giving good results, the cast-in inserts also being practical but having disadvantages.

Cast-in inserts are a poor production proposition in the foundry. They are exceedingly liable to shift on the core when in the mould and cause the bore of the insert to be eccentric with the outside when finish-machined. The cast-in insert of any material is prone to develop defective thermal contact or to even come loose. Due to casting shrinkage the insert initially may be tight in the cylinder-head when both the head and the insert are hot. As a result either of annealing in the course of production or of the gradual annealing that occurs in operation the casting shrinkage is ultimately largely removed. The less the difference is in the coefficients of expansion of the materials of the head and the insert, the greater will be the chance of a cast-in insert remaining tight. Aluminum-bronze cast-in inserts have on this account proved markedly superior to steel or cast iron. Mr. Heron states that cast-in inserts are an example of evading a design issue at the expense of the foundry.

The expanded-in insert gives excellent results in annealed cylinder-heads. The amount of expansion required has, however, to be determined experimentally in each case and is rather difficult to control. Shrunk-in aluminum-bronze inserts have been developed by the engineering division of the U. S. Air Corps and have given excellent results. Shrinking fits and temperatures are much more readily controlled and inspected than either casting-in or expanding. With shrunk seat-inserts the cylinder-head has to be machined partially, the inserts then shrunk in and the head reset for final machining, which is somewhat of a disadvantage. The shrunk insert, at least for inclined valves, is difficult to apply to other than bolted-on or screwed-on heads. It is possible with a shrunk-in insert and with known shrinking allowances and temperature conditions to be certain that good thermal contact always exists.

The greater conductivity of bronze in comparison with cast iron or steel is a distinct advantages for an insert. While the radial effect is of little consequence, the increase of possible circumferential heat-flow within the insert may in some cases have a marked effect on the valve cooling, particularly where two exhaust-valves are used. In three- or four-valve cylinders it is often noticeable that the points of closest proximity of the exhaust valves are considerably hotter than the most widely separated portions.

Valve Timing.—It is in valve timing that consderable difference of opinion prevails among engineers, and it is rare that one will see the same formula in different motors. It is true that the same timing could not be used with motors of different construction, as there are many factors which determine the amount of lead to be given to the valves. The most important of these is the relative size of the valve to the cylinder bore, the speed of rotation it is desired to obtain, the fuel efficiency, the location and method of operation of the valves, and other factors too numerous to mention.

Most of the readers should now be familiar with the cycle of operation of the internal-combustion motor of the four-stroke type, and it seems unncessary to go into detail except to present a review. The first stroke of the piston is one in which a charge of gas is taken into the motor; the second stroke, which is in reverse direction to the first, is a compression stroke, at the end of which the spark takes place, exploding the charge and driving the piston down on the third or expansion stroke, in which the piston moves in the same direction as in the intake stroke, and finally, after the piston has nearly reached the end of this stroke, another valve opens to allow the burned gases to escape, and remains open until the piston has reached the end of the fourth stroke and is in a position to begin the series over again. The ends of the strokes are reached when the piston comes to a stop at either top or bottom of the cylinder and reverses its motion. That point is known as a center, and there are two for each cylinder, top and bottom centers, respectively.

All circles may be divided into 360 parts, each of which is known as a degree, and, in turn, each of these degrees may be again divided into minutes and seconds, though we need not concern ourselves with anything less than the degree in valve timing. Each stroke of the piston represents 180 degrees travel of the crank, because two strokes represent one complete revolution of 360 degrees. The top and bottom centers are therefore separated by 180 degrees. Theoretically each phase of a four-cycle engine begins and ends at a center, though in actual practice the inertia or movement of the gases makes it necessary to allow a lead or lag to the valve, as the case may be. If a valve opens before a center, the distance is called "lead"; if it closes after a center, this distance is known as "lag." The profile of the cams ordinarily used to open or close the valves represents a considerable time in relation to the 180 degrees of the crankshaft travel, and the area of the passages through which the gases are admitted or exhausted is quite small owing to the necessity of having to open or close the valves at stated times; therefore, to open an adequately large passage for the gases it is necessary to open the valves earlier and close them later than at centers.

Advanced Exhaust Valve Opening Essential.—That advancing the opening of the exhaust valve was of value was discovered on the early motors and is explained by the necessity of releasing a large amount of gas, the volume of which has been greatly raised by the heat of combustion. When the inlet valves are mechanically operated it was found that allowing them to lag in some instances at closing enabled the inspiration of a greater volume of gas. Disregarding the inertia or flow of the gases, opening the exhaust at center would enable one to obtain full value of the expanding gases the entire length of the piston stroke, and it would not be necessary to keep the valve open after the top center, as the reverse stroke would produce a suction effect which might draw some of the inert charge back into the cylinder. On the other hand, giving full consideration to the inertia of the gas, and the fact that piston movement is very little in proportion to angular movement of crankpin at either top or bottom centers, opening the valve before center is reached will provide for quick expulsion of the gases, which have sufficient velocity at the end of the

stroke, so that if the valve is allowed to remain open a little longer, the amount of lag varying with the experience of the designer, the cylinder is cleared in a more thorough manner.

Blowing Back.—When the factor of retarded opening is considered without reckoning the inertia of the gases, it would appear that if the valve were allowed to remain open after center had passed, say, on the closing of the inlet, the piston, having reversed its motion, would have the effect of expelling part of the fresh charge through the still open valve as it passed inward at its compression stroke. This effect is called "blowing back," and is often noted with motors where the valve settings are not absolutely correct, or where the valve springs or seats are defective and prevent proper closing. This factor is not of as much import as might appear, as on closer consideration it will be seen that the movement of the piston as the crank reaches either end of the stroke is less per degree of angular movement than it is when the angle of the connecting rod is greater. Then, again, a certain definite though small period of time is required for the reversal of motion of the piston, during which time the crank is in motion but the piston practically at a standstill. If the valves are allowed to remain open during this period, the passage of the gas in or out of the cylinder will be by its own momentum. There is very little piston movement either up or down for an interval corresponding to about 30 degrees crank travel either side of a top or bottom center position of the crankpin.

Why Lead is Given Exhaust Valve.—The faster a motor turns, all other things being equal, the greater the amount of lead or advance it is necessary to give the opening of the exhaust valve. It is self-evident truth that if the speed of a motor is doubled it travels twice as many degrees in the time necessary to lower the pressure. As most designers are cognizant of this fact, the valves are proportioned accordingly. It is well to consider in this respect that the cam profile has much to do with the manner in which the valve is opened; that is, the lift may be abrupt and the gas allowed to escape in a body, or the opening may be gradual, the gas issuing from the cylinder in thin streams. An analogy may be made with the opening of any bottle which contains liquid highly carbonated. If the cork is removed suddenly the gas escapes with a loud pop, but, on the other hand, if the bottle is uncorked gradually, the gas escapes from the receptacle in thin streams around the cork, and passage of the gases to the air is accomplished without noise. While the second plan is not harsh, it is slower than the former, as must be evident. Authorities seem to agree that the valve followers of high-speed engines do not follow the cam profile absolutely and actual valve action and timing in practice is never the same as a theoretical consideration on a drafting board would indicate.

Exhaust Closing, Inlet Opening.—A point which has been much discussed by engineers is the proper relation of the closing of the exhaust valve and the opening of the inlet. It was formerly thought they should succeed each other, the exhaust closing at upper dead center and the inlet opening immediately afterward. The reason why a certain amount of lag is given the exhaust closing in practice is that the piston cannot drive the gases out of the cylinder unless they are compressed to a degree in excess of that existing in the manifold or passages, and while toward the end of

the stroke this pressure may be feeble, it is nevertheless indispensable. At the end of the piston's stroke, as marked by the upper dead center, this *compression still exists, no matter how little it may be, so that if the exhaust valve is closed and the inlet opened immediately afterward,* the pressure which exists in the cylinder may retard the entrance of the fresh gas and a certain portion of the inert gas may penetrate into the manifold. As the piston immediately begins to aspirate, this may not be serious, but as these gases are drawn back into the cylinder the fresh charge will be diluted and weakened in value. If the sparkplug is in a pocket, the points may be surrounded by this weak gas, and the explosion will not be nearly as energetic as when the ignition spark takes place in pure mixture.

It is a well-known fact that the exhaust valve should close after dead center and that a certain amount of lag should be given to opening of the inlet. The lag given the closing of the exhaust valve need not be as great as that given the closing of the inlet valve. Assuming that the excess pressure of the exhaust will equal the depression during aspiration, the time necessary to complete the emptying of the cylinder will be proportional to the volume of the gas within it. At the end of the suction stroke the volume of gas contained in the cylinder is equal to the cylindrical volume plus the space of the combustion-chamber. At the end of the exhaust stroke the volume is but that of the dead space, and from one-third to one-fifth its volume before compression. While it is natural to assume that this excess of burned gas will escape faster than the fresh gas will enter the cylinder, it will be seen that if the inlet valve were allowed to lag twenty degrees, the exhaust valve lag need not be more than ten degrees, providing that the capacity of the combustion-chamber was such that the gases occupied one-quarter of their former volume. In later forms of engines, in order to overcome the lag in valve operating mechanisms, the inlet valve starts to open before the exhaust is completely closed, though at high speeds, the overlap is not enough to have both valves actually open at the same time. What really occurs is that one valve is practically closed before the other one really opens.

It is evident that no absolute rule can be given, as back pressure will vary with the design of the valve passages, the exhaust rings, stacks or manifolds and the construction of the muffler, if any is used. The more direct the opening, the sooner the valve can be closed and the better the cylinder cleared. Ten degrees represent an appreciable angle of the crank, and the time required for the crank to cover this angular motion is not inconsiderable and an important quantity of the exhaust may escape, but the piston is very close to the dead center after the distance has been covered and has moved but little.

Before the inlet valve opens there should be a certain depression in the cylinder, and considerable lag is sometimes allowed before the depression is appreciable. On high-speed engines, to make sure that the inlet will open in time, it is often given a lead of five degrees and there is an actual overlap between inlet opening and exhaust closing, this being as much as fifteen to twenty degrees. So far as the volume of fresh gas introduced during the admission stroke is concerned, this is determined

by the displacement of the piston between the point where the inlet valve opens and the point of closing, assuming that sufficient gas has been inspired so that an equilibrium of pressure has been established between the interior of the cylinder and the outer air. The point of inlet opening varies with different motors. It would appear that a fair amount of lag would be fifteen or twenty degrees past top center for the inlet opening, as a certain depression will exist in the cylinder, assuming that the exhaust valve has closed ten degrees after center, and at the same time the piston has not gone down far enough on its stroke to materially decrease the amount of gas which will be taken into the cylinder.

Closing the Inlet Valve.—As in the case with the other points of opening and closing, there is a wide diversity of practice as relates to closing the inlet valve. Some of the designers close this exactly at bottom center, but this practice cannot be commended, as there is a considerable portion of time, at least ten or fifteen degrees angular motion of the crank, before the piston will commence to travel to any extent on its compression stroke. The gases rushing into the cylinder have considerable velocity, and unless an equilibrium is obtained between the pressure inside and that of the atmosphere outside, they will continue to rush into the cylinder even after the piston ceases to exert any suction effect.

For this reason, if the inlet valve was closed exactly on center, a full charge would not be inspired into the cylinder, though if the time of closing is delayed, this momentum or inertia of the gas will be enough to insure that a maximum charge is taken into the cylinder. The writer considers that nothing will be gained if the valve is allowed to remain open longer than 25 to 30 degrees, and an analysis of practice in this respect would seem to confirm this opinion. From that point in the crank movement the piston travel increases and the compressive effect is appreciable, and it would appear that a considerable proportion of the charge might be blown back into the manifold and carburetor if the valve were allowed to remain open beyond a point corresponding to 30 degrees angular movement of the crank, though in some modern engines, the inlet closing lag may be as late as 35 or 40 degrees for Vee-type engines and over 70 degrees on radial cylinder engines as measured by crankpin travel.

Time of Ignition.—In this country engineers unite in providing a variable time of ignition, though abroad some difference of opinion is noted on this point. The practice of advancing the time of ignition, when affected electrically, was severely condemned by early engine makers, these maintaining that it was necessary because of insufficient heat and volume of the spark, and it was thought that advancing ignition was injurious. The engineers of today appreciate the fact that the heat of the electric spark, especially when from a mechanical generator of electrical energy, is the only means by which we can obtain practically instantaneous explosion, as required by the operation of motors at high speeds, and for the combustion of large volumes of gas and that firing takes an appreciable, though very small period of time.

It is apparent that a motor with a fixed point of ignition is not as desirable in automobiles or airplanes where the engine is started by "swinging the stick," as one in which the ignition can be advanced to best

meet different requirements of starting and running and the writer does not readily perceive any advantage outside of simplicity of control in establishing a fixed point of ignition for automobile engines. In fact, there seems to be some difference of opinion among those designers who favor fixed ignition, and in one case this is located 43 degrees ahead of center, and in another motor the point is fixed at twenty degrees, so that it may be said that this will vary as much as 100 per cent in various forms. This

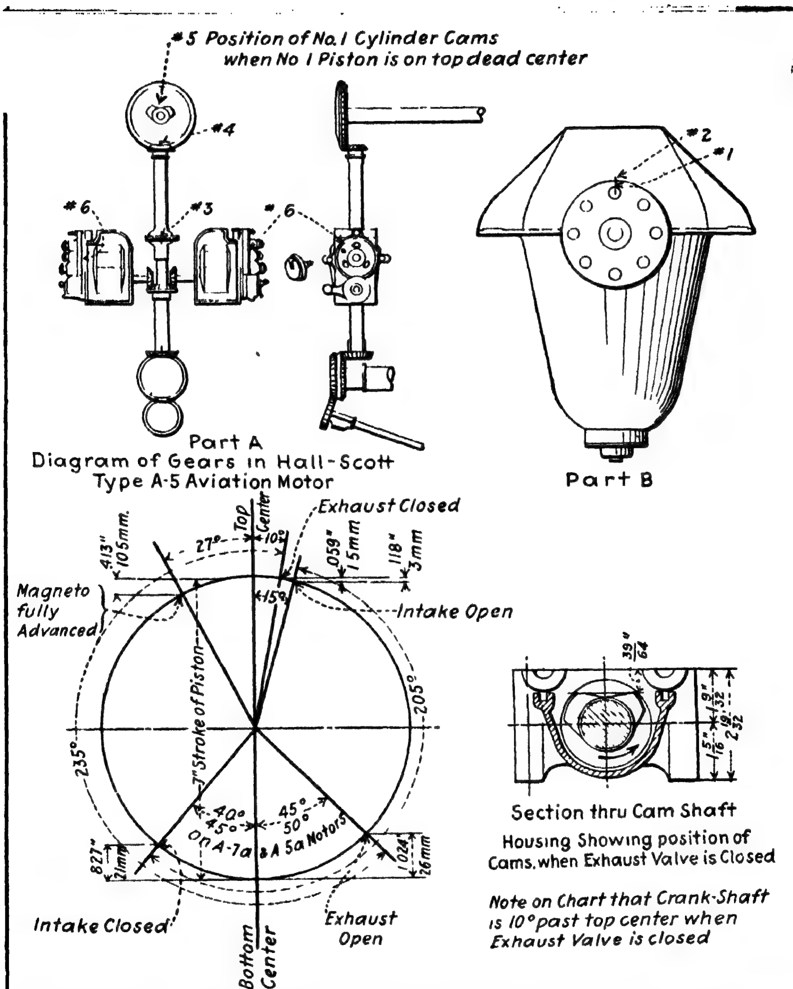


Fig. 301.—Diagrams Explaining Valve and Ignition Timing of Hall-Scott Aviation Engine.

point will vary with different methods of ignition, as well as the location of the sparkplugs. To secure simplicity, some airplane engines use set spark; if an advancing and retarding mechanism is fitted, it is only to facilitate starting, as the spark is kept advanced while in flight, and control is by throttle alone. As the spark should be retarded when an engine is started by turn-

ing a propeller by hand, to prevent danger of "kick back" and injury to person starting the motor, variable spark timing is still desirable aviation practice.

No Set Rules for Valve Timing.—It is obvious by consideration of the foregoing that there can be no arbitrary rules established for timing, because of the many conditions which determine the best times for opening and closing the valves. It is customary to try various settings when a new motor is built and given brake tests until the most satisfactory points are determined, and the setting which will be very suitable for one motor

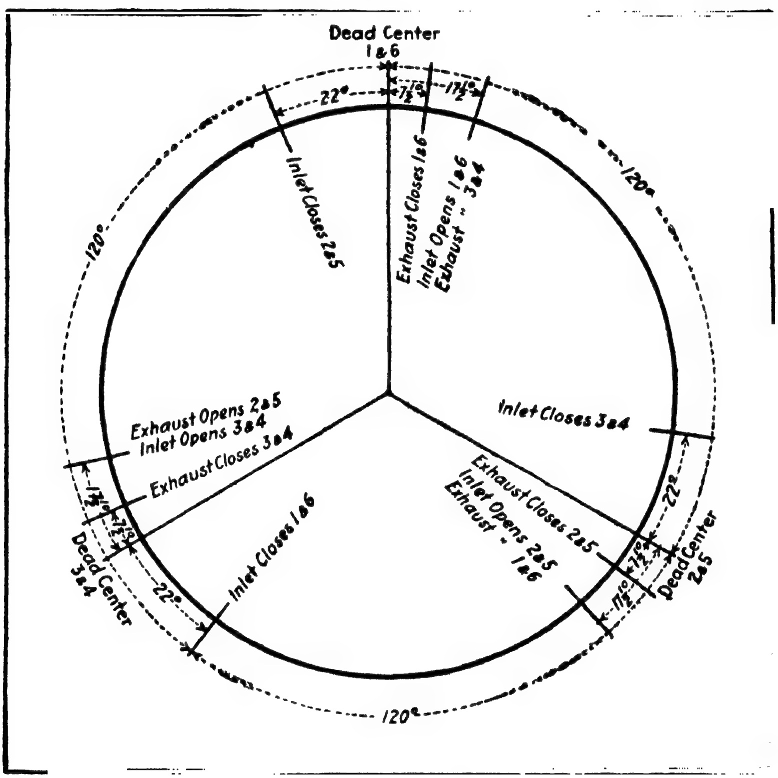


Fig. 302.—Typical Timing Disc Lay-Out for Six-Cylinder Automotive Engine.

is seldom right for one of different design. The timing diagram shown at Fig. 301 applies to the early Hall-Scott engine, and may be considered typical. It should be easily followed in view of the very complete explanation given in preceding pages. Another six-cylinder engine diagram is shown at Fig. 302, and an eight-cylinder timing diagram is shown at Fig. 303. In timing automobile engines no trouble is experienced, because timing marks are nearly always indicated on the engine fly-wheel and these register with an indicating trammel on the crankcase. To time an airplane engine accurately which has no fly-wheel as is necessary to test for a suspected camshaft defect, a timing disc of aluminum is attached to the crankshaft which has the timing marks indicated thereon. If the disc is

made ten or twelve inches in diameter, it may be divided into degrees without difficulty and sufficient interval will exist between graduations to read the angular movement accurately.

Valve timing markings depend entirely upon the design of the motor to be timed. If a disc is graduated in degrees it may be used to time any motor though discs may be marked in such a way that they can be used only with certain engines for which they were designed. As an example of how timing varies, the following tabulation shows that of two representative water-cooled Vee engines and two air-cooled radial engines.

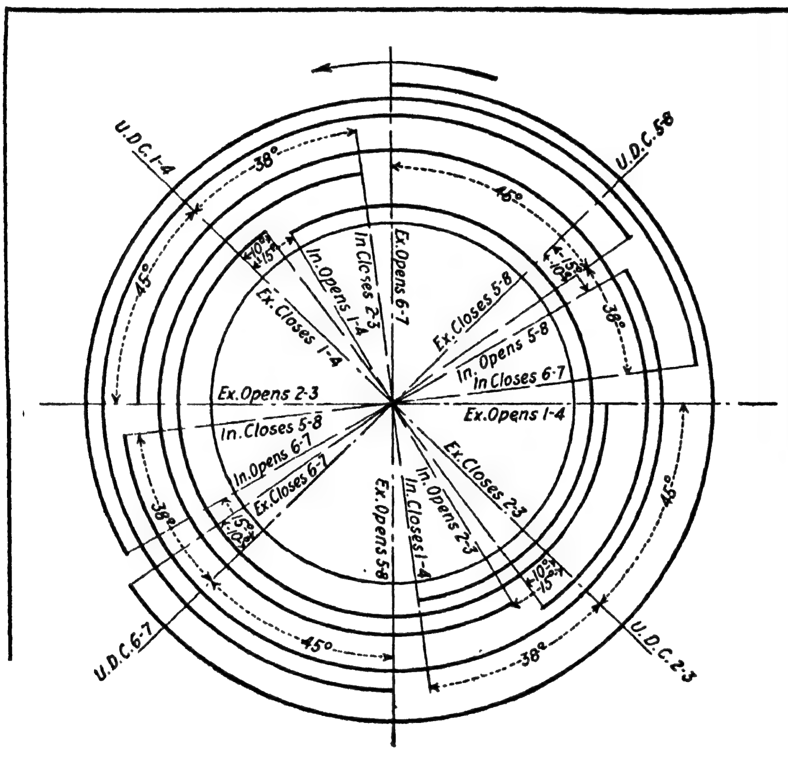


Fig. 303.—Timing Diagram of Typical Eight-Cylinder "Vee" Engine.

Magneto timing of Curtiss D12 is 32 degrees advance for left magneto and 36 degrees advance for right magneto. Magneto full advance on Packard 2A 2,500 is 44 degrees ahead of T. C. for 5.5 to 1 compression ratio and 50 degrees for 5 to 1 compression ratio. The Wasp magneto is timed 30 degrees advanced, the Wright J5 is also timed 30 degrees advanced.

How an Engine is Timed.—In timing a motor from the marks on the timing disc rim it is necessary to regulate the valves of but one cylinder at a time. Assuming that the disc is revolving in the direction of engine rotation, and that we are timing a simple four-cylinder in-line motor

having the firing order of the cylinders 1-3-4-2, the operation of timing would be carried on as follows: The crankshaft would be revolved until the line marked "Exhaust opens 1 and 4" registered with the trammel point or index mark on the motor bed. At this point the exhaust valve of either cylinder No. 1 or No. 4 should begin to open. This can be easily determined by noting which of these cylinders holds the compressed charge ready for ignition. Assuming that the spark has occurred in cylinder No. 1, then when the timing disc is turned from the position to that in which the line marked "Exhaust opens 1 and 4" coincides with trammel point, the valve plunger under the exhaust-valve rocker of cylinder No. 1 should be adjusted in such a way that there is no clearance between the rocker

VALVE TIMING OF TYPICAL AMERICAN AVIATION ENGINES

Make of Engine	Inlet Opens	Inlet Closes	Exhaust Opens	Exhaust Closes
Curtiss D12	5° Before T C	35° After B C.	55° Before B C.	10° After T.C.
Packard 2A 2500	10° After T C	45° After B C	48° Before B.C	8° After T.C.
Wright J5	8° Before T.C.	60° After B C	60° Before B.C	8° After T.C.
Pratt and Whitney Wasp	26° Before T.C	76° After B C.	71° Before B.C.	31° After T.C.

and the valve stem. Further movement of the disc in the same direction should produce opening of the exhaust valve. The disc is turned about 225 degrees, or a little less than three-quarters of a revolution; then the line marked "Exhaust closes 1 and 4" will register with the trammel point. At this period the valve-rocker and the valve-stem should separate and a certain amount of clearance obtain between them as indicated in the instructions of the engine builder. The next cylinder to time would be No. 3. The crankshaft is rotated until mark "Exhaust opens 2 and 3" comes in line with the trammel. At this point the exhaust valve of cylinder No. 3 should be just about opening. The closing is determined by rotating the shaft until the line "Exhaust closes 2 and 3" comes under the trammel. This operation is carried on with all the cylinders, it being well to remember that but one cylinder is working at a time and that a half-revolution of the timing disc corresponds to a full working stroke of all the cylinders, and that while one is exhausting the others are respectively taking in a new charge, compressing and exploding. For instance, if cylinder No. 1 has just completed its power-stroke, the piston in cylinder No. 3 has reached the point where the gas may be ignited to advantage. The piston of cylinder No. 4, which is third to fire, is at the bottom of its stroke, and will have inspired a charge, while cylinder No. 2, which is the last to fire, will have just finished expelling a charge of burned gas, and will be starting the intake stroke. This timing relates to a four-cylinder engine in order to simplify the explanation. The timing instructions given apply only to the conventional motor types.

Timing Gnome Rotary Engines.—Rotary cylinder engines, especially the Gnome "monosoupape," have a distinctive valve timing on account of the peculiarities of design. In the design of the early Gnome motor, a

cycle of operations somewhat different from that employed in the ordinary four-cycle engine is made use of. This cycle does away with the need for the usual inlet valve and makes the engine operable with only a single valve, hence the name *monosoupape,* or "single-valve." The cycle is as follows: A charge being compressed in the outer end of the cylinder or combustion-chamber, it is ignited by a spark produced by the sparkplug located in the side of this chamber, and the burning charge expands as the piston moves down in the cylinder while the latter revolves around the crankshaft. When the piston is about half-way down on the power stroke, the exhaust valve, which is located in the center of the cylinder-head, is mechanically opened, and during the following upstroke of the piston the burnt gases are expelled from the cylinder through the exhaust valve directly into the atmosphere.

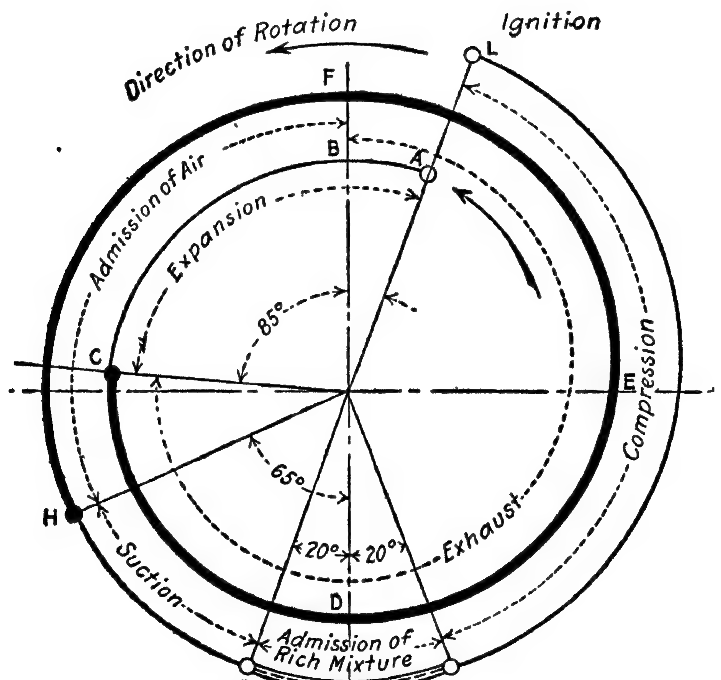


Fig. 304.—Timing Diagram Showing Unusual Valve Timing of Gnome "Monosoupape" Rotary Motor, now an Obsolete Design.

Instead of closing at the end of the exhaust stroke, or a few degrees thereafter, the exhaust valve is held open for about two-thirds of the following inlet stroke of the piston, with the result that fresh air is drawn through the exhaust valve into the cylinder. When the cylinder is still 65 degrees from the end of the inlet half-revolution, the exhaust valve

closes. As no more air can get into the cylinder, and as the piston continues to move inwardly, it is obvious that a partial vacuum is formed.

When the cylinder approaches within twenty degrees of the end of the inlet half-revolution a series of small inlet ports all around the circumference of the cylinder wall is uncovered by the top edge of the piston, whereby the combustion-chamber is placed in communication with the crank chamber. As the pressure in the crank chamber is substantially atmospheric and that in the combustion-chamber is below atmospheric, there results a suction effect which causes the air from the crank chamber to flow into the combustion-chamber. The air in the crank chamber is heavily charged with gasoline vapor, which is due to the fact that a spray nozzle connected with the gasoline supply tank is located inside the chamber. The proportion of gasoline vapor in the air in the crank chamber is several times as great as in the ordinary combustible mixture drawn from a carburetor into the cylinder. This extra-rich mixture is diluted in the combustion-chamber with the air which entered it through the exhaust valve during the first part of the inlet stroke, thus forming a mixture of the proper proportion for complete combustion.

The inlet ports in the cylinder wall remain open until twenty degrees of the compression half-revolution has been completed, and from that moment to near the end of the compression stroke the gases are compressed in the cylinder. Near the end of the stroke ignition takes place and this completes the cycle. The exact timing of the different phases of the cycle is shown in the diagram at Fig. 304. It will be seen that ignition occurs substantially twenty degrees ahead of the outer dead center, and expansion of the burning gases continues until 85 degrees past the outer dead center, when the piston is a little past half-stroke. Then the exhaust valve opens and remains open for somewhat more than a complete revolution of the cylinders, or, to be exact, for 390 degrees of cylinder travel, until 115 degrees past the top dead center on the second revolution. Then for 45 degrees of travel the charge within the cylinder is expanded, whereupon the inlet ports are uncovered and remain open for 40 degrees of cylinder travel, twenty degrees on each side of the inward dead center position. The reader should bear in mind that this unconventional timing is a result of the design and that the Gnome engine is not used on modern airplanes because of its high fuel and oil consumption.

Finishing Cylinder Bores.—A tool for finishing cylinder bores which has been adapted from the service into the production field and which has achieved a considerable degree of popularity in this latter field in a short time, is the Hutto grinder, manufactured by the Hutto Engineering Co., Detroit, Mich., recently described and illustrated in *Automotive Industries* magazine. In the production field this tool was pioneered by the Continental Motors Corp., and it has since been adopted by such companies as Hupp, Hudson-Essex, Buick, Cadillac, Chrysler, Chevrolet and Franklin in this country and Daimler in England. It is also used by aircraft engine builders.

In the service field the Hutto grinder was introduced as a three-stone tool for use with a portable drill, but following the earlier experiments in production work it has been developed into a six-stone tool, which latter

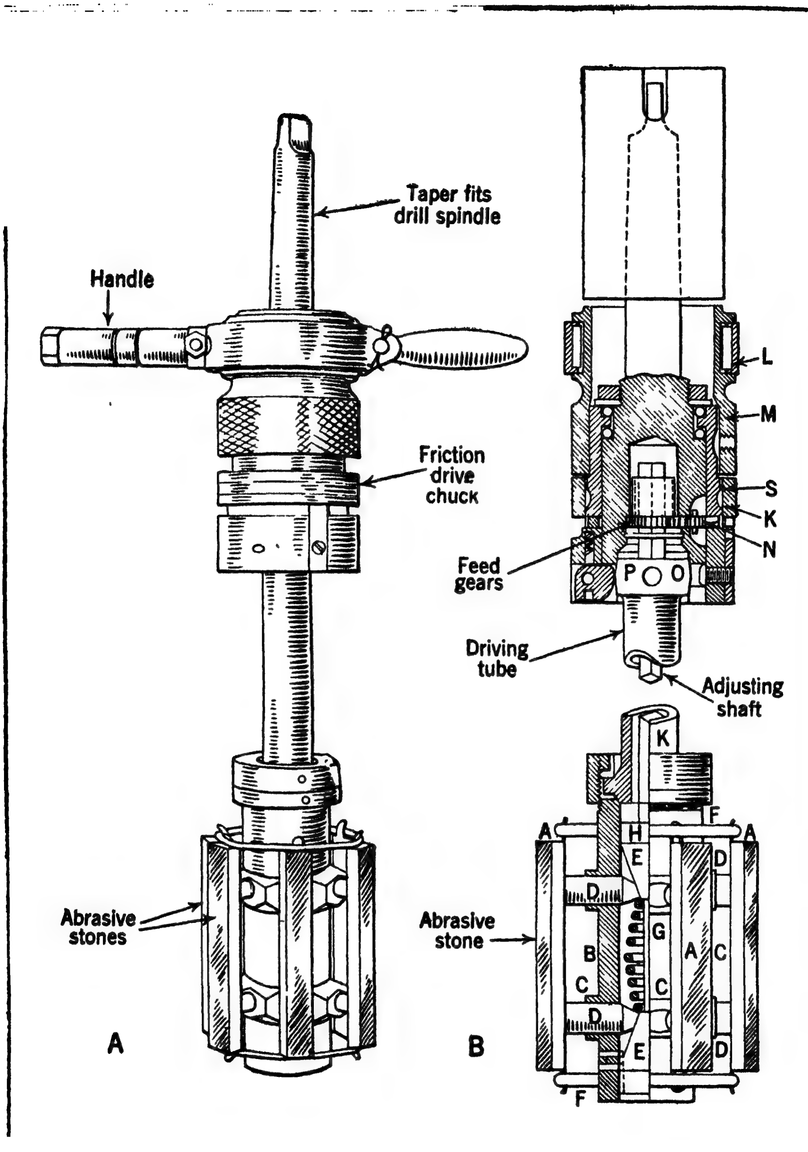


Fig. 305.—Hutto Cylinder Grinder and Friction Type Driving Head Shown at A. The Floating Driving Bar which Encloses a Square Adjusting Shaft and Driving the Grinder Head is Shown in Sectional View at B.

forms the subject of the following description. This improved tool is widely used also in the service field.

As shown in Fig. 305 B, six stones A are mounted on a liberally proportioned, hardened steel body B. For the usual operations on the cylinders of small engines, these abrasive stones are four inches long and $\frac{7}{16}$ inch wide. Each stone is mounted in a pressed steel carrier C, being se-

cured by die-casting white metal in the space between the stone and the sides of the pressed steel member. Two ground pins D, D are riveted into the pressed steel carrier C, and their opposite ends are first turned cone-shaped and then beveled off to the same angle as the interior adjusting cones E, E. After the pins have been riveted in place, this beveled surface is ground to the desired dimension from the stone mounting surface.

Pilot holes for the carrier pins C are ground to dimension and accurate location after the body B of the grinder has been hardened. As the stones are formed with care as to thickness and parallelism of the inner and outer surfaces, the working relationship of the cutting face of the stone and the bevel bearing surfaces at the inner ends of the pins are established within close commercial limits. The assembly of pins and pressed steel carrier is made in a fixture which insures parallelism and accurate spacing of the pins. As each stone is assembled, it provides for a working wear of one-eighth inch. Regardless of the size of the bore in which the grinder is to function, the cutting face of the stone has a radius of 1⅛ inches, when viewed from the end. This round form has supplanted the original flat face as the flat stones showed some tendency to chatter due to the high pressures developed at the corners. As illustrated, the stones are at all times drawn toward the center by the endless coiled springs F, F, which are hooked over the ends of all the pressed steel carriers.

The outstanding feature of this grinder head is the positive adjustment of the abrasive stones. The stones are advanced with no intermediate spring action which might tend to introduce bell mouthing at the end of the bore. As shown in the cross section, the beveled inner ends of the carrier pins D, D, bear on opposed cones E, E. Neither of these cones is fixed in the body of the grinder. A shank G of one-quarter inch diameter passes through the upper cone and is threaded into the lower one, with a collar H at the back of the upper cone. This arrangement is similar to the double cone piston turning clutch which is used very largely in the industry. The lower cone is restrained from rotation in the body by means of two pins which engage in a slot as shown at the lower left. With this arrangement a barrel type coil spring I separates the cones to the extent allowed by the screw adjustment described in the previous paragraph, but both cones float and are positioned by the reaction pressure developed at the carrier pins. The originator of this grinder regards this floating cone arrangement as the foundation of its success, as it allows for equalization of pressure and position of the abrasive stones and even corrects for differences in hardness at the opposite ends of the stones. In keeping with the character of the unit, the adjusting cones are hardened and ground. In the usual production arrangement, the adjusting cones, or, rather, the shank or spindle on which they are mounted, is controlled by a square shaft J which passes up through the floating bar K which drives the grinder. The lower end of this square shaft J fits freely into a square socket in the collared head at the upper end of the adjusting spindle. The upper end of this square control shaft is fitted into one of the several adjusting means which are provided on the various forms of drivers.

In operation, the adjusting cones are spread apart both before and after each operation. In other words, the grinder is introduced into the cylinder barrel before being set to a cutting position, and is contracted before being removed from the cylinder at the end of the grinding operation. In the usual practice, the cylinder block is set on a reciprocating table, while the grinder heads rotate in one plane. During the loading and unloading period the grinder heads are contracted and raised into pilot sleeves which are practically concentric with the cylinder bores, and the heads are returned to the working position within the cylinder bores before actual operation is started again. Experience has demonstrated that the stones should over-run the ends of the bores by three-quarters to one inch and should completely over-run the middle point of the bore. That is the outer end of each stone should cross the horizontal center of the bore on each stroke. This point is highly essential to the production of straight bores of uniform size, and can be attained by varying either the length of stroke of the reciprocating table or the length of the grinder stones. However, in practically all cases involving engines of ordinary size, stones of four inch length will meet the situation.

Best Speed of Rotation of Grinder Head.—For the cylinder of ordinary size, best results are obtained with about 350 r.p.m. of the grinder heads and 75 reciprocation strokes per minute. It is stated that with these speeds a finish superior to that of the present will be obtained and the abrasive stones will wear about three times as long, with almost complete elimination of glazing. With abrasive stones of the correct grain and grit for iron of a given hardness, glazing can be charged to the slow speed of operation. This is a condition much like that of chip clearance. When the speed is too low the abraded material piles up on the stone instead of being cut free. Another good result of these speeds of rotation and reciprocation is a grinder pattern in which the intersecting lines cross at almost right angles in a diamond-shaped arrangement. The diagonals of these diamonds are in the vertical and horizontal planes and the vertical angles are slightly smaller than the horizontal.

In this connection the manufacturer states that the grinder marks should be discernible but not prominent, to insure the best commercial finish known as a satin finish. Recently one of the well-known companies had a lot of twenty cylinders put through with a mirror finish which can be produced with practically the same ease as the commercial finish. In about two weeks' time, this production department called back and cancelled any further mirror finish, as it was almost impossible to get the rings to wear in to a seal. The commercial or satin finish has been found to be one of the happy compromises which are so frequent in automotive production. At the end of the running-in period, the cylinder shows no wear but a good finish, with rings properly seated.

Kerosene is the lubricant or coolant used, and should be supplied in liberal volume. Several qualities of abrasive stones are carried in stock constantly, and the choice for the best results is governed by the hardness of the cylinder iron, the amount of stock to be removed, the character of finish desired and the speed of the operation as regards reciprocation and rotation. As a general rule, the hardness of the stones is inversely propor-

tional to the hardness of the cylinder iron. The usual finish allowance ranges from .001 inch to .0015 inch for one grinding operation which is being performed in many plants in less than one minute, floor to floor time. Hupp utilizes a semi-finishing grinding operation with rather coarse stones, starting with a finish allowance of .003 inch and allowing a maximum of .0005 inch for the final satin finishing operation.

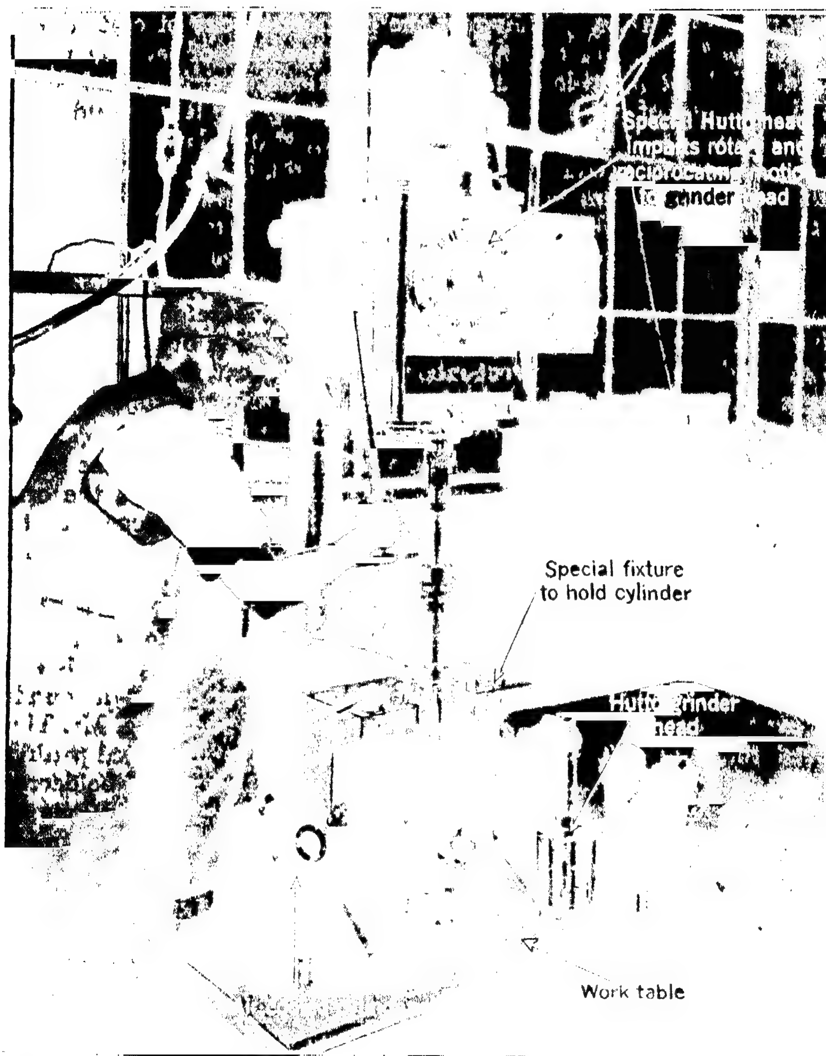


Fig. 305A.—The Hutto Cylinder Grinder in Use at the Factory of the Wright Aeronautical Corporation Finishing "Whirlwind" Engine Cylinder Bores.

Advantages of Honing and Lapping Cylinder Bores.—The comparative advantages of grinding, reaming, honing and lapping cylinder bores in producing the most desirable finish resulted in an instructive meeting replete with keen discussion at a session of the Detroit Section of the S.A.E.

The Chrysler Corporation, said C. A. Bolus, decided, after a period of experimentation, that lapping would produce at lower cost a better finish than had been obtained by grinding, and would have the additional advantage of avoiding the necessity for keeping skilled operators on the finishing operation. For a time, grinding of the cylinder-blocks of certain engines was continued but, a few months later, the practice of lapping was fully adopted. Cylinders are lapped in two operations, roughing and finishing, he said, the type of lap being the same except for the length and shape of the stones on the cutting surfaces. Roughing stones have a concave surface, finishing stones, convex. The quality of the latter is also finer. The amount of material removed depends on the condition of the reamed surface and varies from 0.0005 to 0.0015 inch. The surface must be smooth, within close limits as to out-of-roundness, and without taper. Experience has proved, continued Mr. Bolus, that engines with lapped cylinders leave the running-in stands with the bores in a very satisfactory condition. As the investment in lapping equipment is only a fraction of that required for grinding and the saving in direct labor cost is so great, he believed that the changes in the methods of machining that were effected constitute one of the outstanding contributions toward economy in production.

The Wilson Foundry & Machine Co., said F. N. Thiefels, has been lapping cylinder bores continually since 1921 and still employs the original method. Not so much attention is paid to the lapping process as to the preparation of the bores before lapping. Lapping then becomes merely smoothing, during which approximately 0.0005 inch is removed and for which from fifteen to twenty seconds is required. Bores that exceed 0.0005 inch in out-of-roundness or 0.0005 inch in taper are salvaged on a single-spindle vertical drilling-machine in which lap is used that expands equally on all hones and removes a certain percentage of taper or out-of-roundness. After experimenting with various other types of lap and hone, Mr. Thiefels believed that the cost of lapping would be greatly increased with their adoption and that the accuracy of the boring operation would be decreased. A lapping machine of the company's own design is employed, one of the features of which is the use of a heart-shaped or constant-speed cam for the stroke of the head that carries the laps. It is said to be superior to a crank motion, for it eliminates the dwell at the end of the stroke that has a tendency to bell-mouth the bores. Approximately 250 cylinder-blocks can be lapped on each set of hones at a cost, including labor, of five cents per block. Knight engine sleeves require from 30 to 45 seconds to lap, depending on the amount of stock to be removed. From 0.0005 to 0.0010 inch is allowed. The cost of lapping is approximately 1½ cents per sleeve or a combined engine cost of fifteen cents for twelve sleeves.

A. H. Fors, of the Continental Motors Corporation, explained in detail the disadvantages that had been found with grinding and traced the steps that led to the adoption of the method of lapping at present in use. In finishing the cylinder-blocks, two boring and one reaming operation bring

the bore to within from 0.015 to 0.018 inch of the final size. The size-reaming operation is performed on a single-spindle machine so that all the holes in the same block will be of uniform size. Important points in obtaining a correctly reamed bore are the grinding of the reamer and its hardness. Blades of 77-82 scleroscope-hardness are used. For the final finishing or lapping 0.0015 inch is left.

Lapping is performed on a single-spindle machine having a speed of 400 r.p.m. and a hand feed, although machine feed is considered preferable and probably will soon be substituted. In the last two years, several varieties of lap have been tried, but that now in use is made by the Hutto Company and is of the six-stone type embodying a double three-point-contact support for the stone that makes the grinding element self-aligning as well as self-centering. These features are due to the fact that the two adjusting-cones and stone-holder pins that rest upon them have a limited freedom in which to float until the cutting-surfaces of the stones are all parallel. This floating of the adjusting-mechanism also compensates for any unevenness of wear of the stones because of lack of uniformity in their composition. A six-cylinder block can be reamed in 5.6 minutes and lapped in 9.0 minutes, a total of 14.6 minutes. Grinding a cylinder-block of the same size required 54 minutes. Mr. Fors believes that lapping produces a better and more accurate cylinder bore than does grinding, and that the labor cost is less.

H. C. Miller, of the Oakland Motor Car Co., stated that the difficulty his firm had encountered because of the bell-mouthing of the bore at the top had been overcome by placing a scrap block above the cylinder being honed; this prevents the hones from flying out. Another difficulty, caused by the laps' wearing and the slots' becoming larger, and the consequent backing-up of the stones against one side, was turned into an advantage. The stones are turned round as soon as they get half-way to a full round surface and a narrow cutting-edge is produced.

QUESTIONS FOR REVIEW

1. Why is it necessary to provide adequate cooling for the exhaust valves of air-cooled engines?
2. Describe methods of valve cooling.
3. What salts are used for filling valve stems?
4. What materials are used for valve stem guides?
5. What material is used for valve seat inserts in aluminum alloy heads and why?
6. In timing, why is exhaust valve given a lead?
7. What is lag in timing and why is it allowed?
8. What is the time of ignition?
9. Why was Gnome monosoupape motor timing different than the usual method?
10. Describe honing method of finishing cylinder bores.

CHAPTER XXII

AIRCRAFT ENGINE PISTONS, RINGS AND CONNECTING RODS

Construction of Pistons—Wrist Pin Retention Methods—Aluminum Alloy Pistons—Modern Pistons Well Developed—Some Problems in Piston Design—Magnesium Pistons—Aluminum Alloy Piston Forms—Smooth Finish Important—Causes of Piston Slap—Excessive Oil Consumption—Aluminum Pistons Run Cooler—Advantages of High Heat Conductivity—Split Skirt Pistons—Strut Type Piston—Locking Wristpin in Alloy Pistons—Slipper Type Pistons—Slipper Pistons Increase Efficiency—Factors Affecting Clearance—Cylinder Bore and Piston Finish Important—Effect of Finishing Pistons—Dycer-Austin Alloy Piston—Durator Iron Piston—Piston Ring Construction—Concentric vs. Eccentric Rings—Gray Iron Best for Rings—Rings Made from Individual Castings—Reason for Peening Ring Interior—Piston Ring Width—Light Test for Piston Rings—Piston Ring Joints—Oil Rings—Quick Seating Rings—Machining Ring Grooves—Leakproof Piston Rings—Compound and Unusual Piston Rings—Keeping Oil Out of Combustion Chamber—Connecting Rod Forms—Connecting Rod Sections—Connecting Rods for Vee Engines—Rods for W and X Engines—Rods for Radial Cylinder Engines—Ball and Roller Bearing Rods—Roller Separators Important—Method of Using Standard Bearings—Split Connecting Rod Big Ends.

The piston is one of the most important parts of the gasoline motor inasmuch as it is the reciprocating member that receives the impact of the explosion and which transforms the power obtained by the combustion of gas to mechanical motion by means of the connecting rod to which it is attached. The piston is one of the simplest elements of the motor, and it is one component which does not vary much in form in different types of motors. The piston is a cylindrical member provided with a series of grooves in which packing rings are placed on the outside and two bosses which serve to hold the wristpin in its interior. It is usually made of cast iron or aluminum, though in some motors it may be made of steel combined with aluminum, the former material being used at the skirt and the piston being a composite type, having an aluminum head. The use of the more resisting material enables the engineer to use lighter sections where it is important that the weight of this member be kept as low as possible consistent with strength.

Construction of Pistons.—A number of automotive engine piston types are shown at Fig. 306. That at A has a round top and is provided with four split packing rings and two oil grooves. A piston of this type is generally employed in motors where the combustion-chamber is large and where it is desired to obtain a higher degree of compression than would be possible with a flat top piston. This construction is also stronger because of the arched piston top. The most common form of cast-iron piston is that shown at B, and it differs from that previously described only in that it has a flat top. The piston outlined in section at C is a type used on some of the sleeve-valve motors of the Knight pattern, also some aviation engines and has a concave head instead of the convex form shown at A. The design shown at D in side and plan views is the conventional form employed in two-cycle engines. The deflector plate on the top of the cylinder is cast integral and is utilized to prevent the incoming fresh gases from flow-

ing directly over the piston top and out of the exhaust port, which is usually opposite the inlet opening. On the types of two-cycle engines where a two-diameter cylinder is employed, the piston shown at E is used. This is known as a "differential piston," and has an enlarged portion at its lower end which fits the pumping cylinder. The usual form of deflector plate is provided at the top of the piston and one may consider it as two pistons in one.

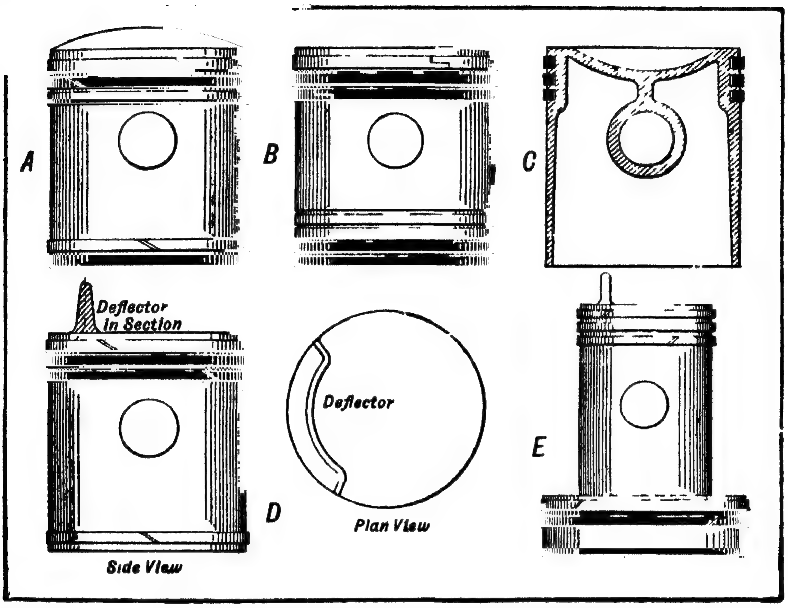


Fig. 306.—Forms of Pistons Generally Employed in Gasoline Engines. A—Dome Head Piston and Three Packing Rings. B—Flat Top Form Almost Universally Used. C—Concave Piston Utilized in Knight Motors and Some Having Overhead Valves. D—Two Cycle Engine Piston with Deflector Plate Cast Integrally. E—Differential or Two Diameter Step Pistons Used in Some Engines Operated on Two-Cycle Principle.

Wristpin Retention.—One of the important conditions in automotive engine piston design is the method of securing the wristpin which is used to connect the piston to the upper end of the connecting rod. Various methods have been devised to keep the pin in place, the most common of these that have received application in automobile and aviation engines being shown at Fig. 307. The wristpin should be retained by some positive means which is not liable to become loose under the vibratory stresses which obtain at this point and in aviation engines, the locking member should be light in weight. If the wristpin was free to move it would work out of the bosses enough so that the end would bear against the cylinder wall. As it is usually made of hardened alloy steel, which is a harder material than cast iron or steel used in cylinder construction, the rubbing action would tend to cut a groove in the cylinder wall which would make for loss of power because it would permit escape of gas. The wristpin member is a simple cylindrical element that fits the bosses closely, and it

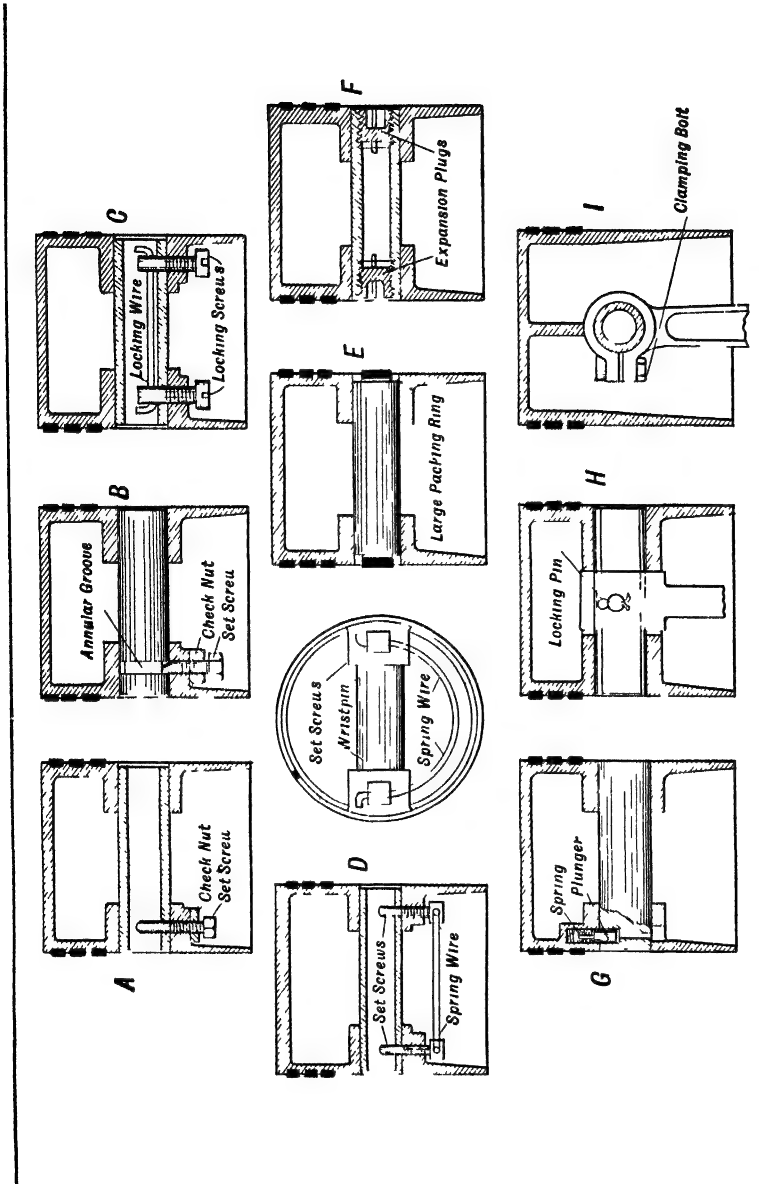


Fig. 307.—Typical Methods of Piston Pin Retention that Have Been Used in Gasoline Engines of American Design. A—Single Set Screw and Lock Nut. B—Set Screw Fitting Groove in Wristpin. C and D—Two Locking Screws Fastened Into the Interior of a Hollow Wristpin. E—Split Ring Holds Pin in Place. F—Use of Tapered Expanding Plugs Outlined. G—Spring Pressed Plunger Type. H—Piston Pin Pinned to Connecting Rod. I—Wristpin Clamped in Connecting Rod Small End by a Bolt.

is nearly always hollow stock A typical piston and connecting rod assembly of early design suitable for conventional four- and six-cylinder-in-line motors, which shows a piston in section, also is presented at Fig. 308. The piston of the early Sturtevant aeronautical motor is shown at Fig. 309, the aluminum piston of the early Thomas airplane motor with piston rings in place is shown at Fig. 310 Note the use of wide piston rings following the

then current automobile practice. Modern engines have narrow rings rather than wide ones. A good view of the wristpin and connecting rod are also given. The semisteel piston of the Gnome "Monosoupape" airplane engine and the unconventional connecting rod assembly are clearly depicted at Fig. 311.

The method of retention shown at A is the simplest and consists of a set screw having a projecting portion passing into the wristpin and holding it in place. The screw is kept from turning or loosening by means of a check nut. The method outlined at B is similar to that shown at A, except

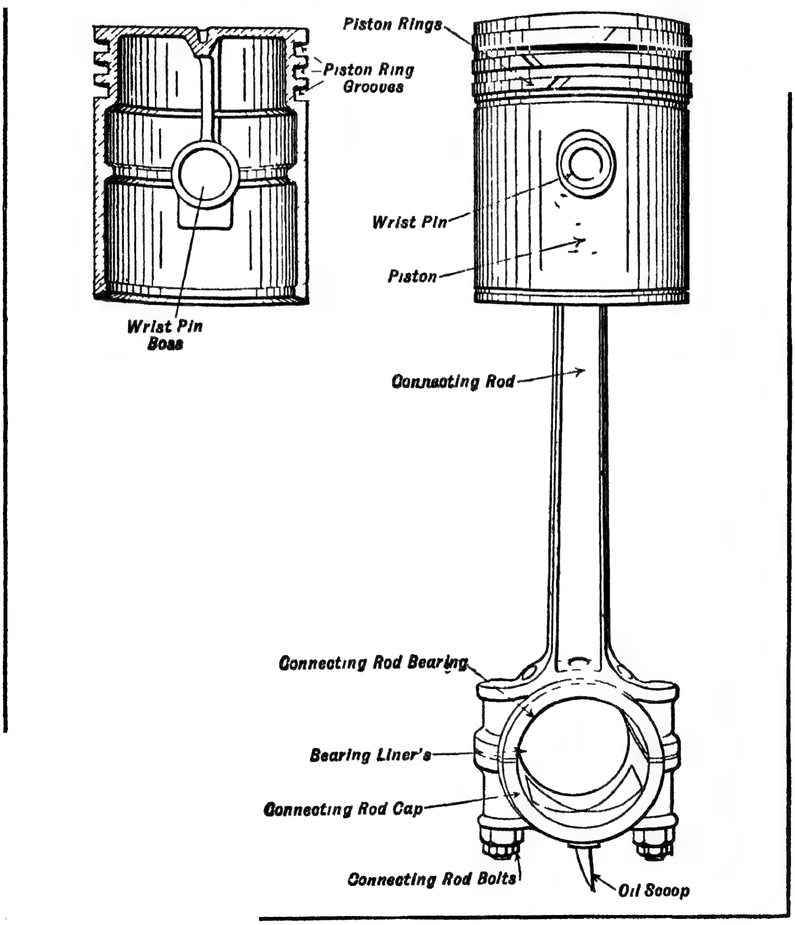


Fig. 308.—Typical Automotive Engine Piston and Connecting Rod Assembly.

that the wristpin is solid and the point of the set screw engages an annular groove turned in the pin for its reception. A very positive method is shown at C. Here the retention screws pass into the wristpin and are then locked by a piece of steel wire which passes through suitable holes in the ends.

The method outlined at D is sometimes employed, and it varies from that shown at C only in that the locking wire, which is made of spring steel, is passed through the heads of the locking screws. Some designers of early engines machined a large groove around the piston at such a point that after the wristpin was put in place a large packing ring was sprung in the groove and utilized to hold the wristpin in place. This method would not be suitable on modern high-speed engines.

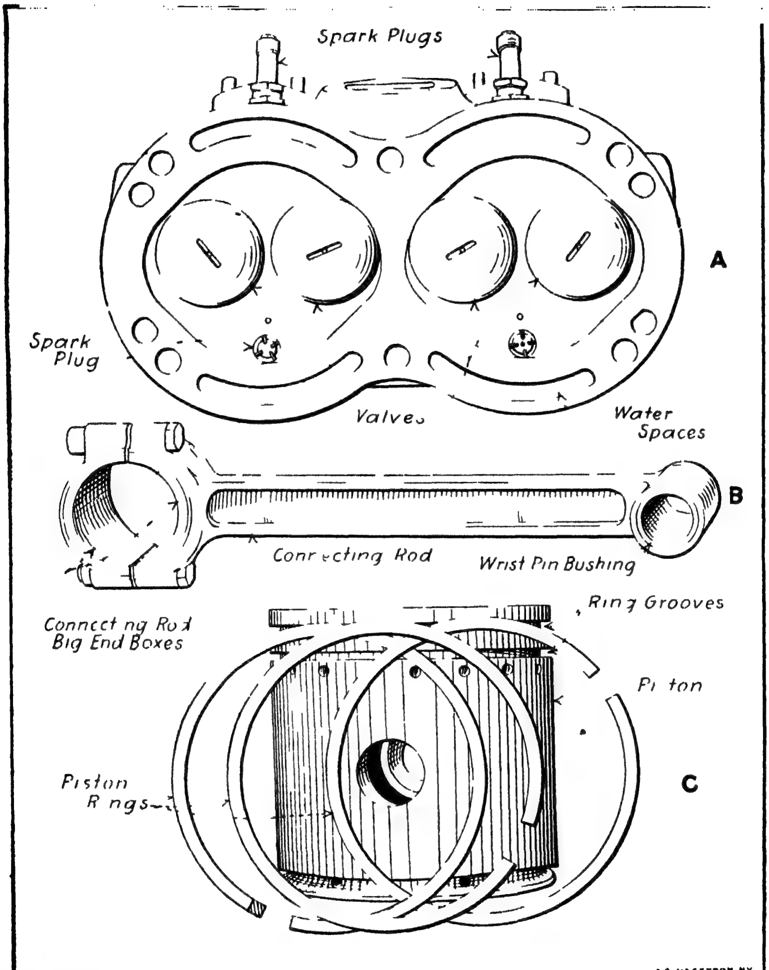


Fig. 309.—Parts of Early Sturtevant Aviation Engine. A—Cylinder Head, Showing Valves. B—Connecting Rod. C—Piston and Rings.

The system shown at F is not so widely used as the simpler methods, because it is more costly and does not offer any greater security when the parts are new than the simple lock shown at A. In this a hollow wristpin is used, having a tapered thread cut at each end. The wristpin is slotted at three or four points, for a distance equal to the length of the boss, and

when taper expansion plugs are screwed in place the ends of the wristpin are expanded against the bosses. This method has the advantage of providing a certain degree of adjustment if the wristpin should loosen up after it has been in use for some time. The taper plugs would be screwed in deeper and the ends of the wristpin expanded proportionately to take up the lost motion. Its greater weight than the simpler system would be a great disadvantage in airplane engines. It is shown merely as a matter of general interest. The method shown at G is an ingenious one. One of the piston bosses is provided with a projection which is drilled out to receive a plunger. The wristpin is provided with a hole of sufficient size to

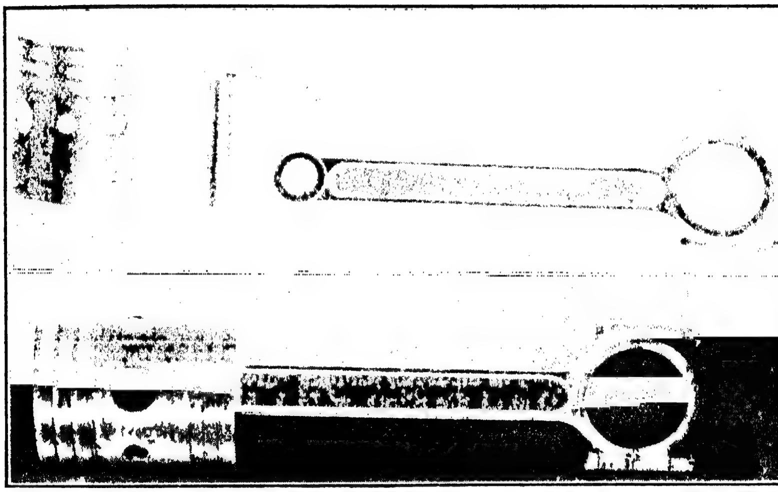


Fig. 310.—Aluminum Piston and Forged Steel Connecting Rod of Early Aviation Engine. Note the Use of Wide Piston Rings at that Time.

receive the plunger, which is kept in place by means of a spring in back of it. This makes a very positive lock and one that can be easily loosened when it is desired to remove the wristpin. To unlock, a piece of fine rod is thrust into the hole at the bottom of the boss which pushes the plunger back against the spring until the wristpin can be pushed out of the piston.

Many engineers think it advisable to oscillate the wristpin in the piston bosses, instead of in the connecting rod small end while others have the wristpin free to move in both the piston and connecting rod end bosses to form a "floating" construction which is said to more evenly distribute the wear on the pin surface. It is argued that this construction gives more bearing surface at the wristpin and also provides for more strength because of the longer bosses that can be used. When the clamp system is followed the piston pin may be held in place by locking it to the connecting rod by some means. At H the simplest method is outlined. This consisted of driving a taper pin through both rod and wristpin and then preventing it from backing out by putting a split cotter through the small end of the tapered locking pin. This has the disadvantage of greatly weakening the wristpin. A better construction is to drive the taper pin through a groove cut into

the top of the wristpin only as the groove does not detract from the strength as much as the hole would. Another method, which is depicted at I, consists of clamping the wristpin by means of a suitable bolt which brings the slit connecting rod end together as shown. This is sometimes used in modern practice where it is considered desirable to lock the wristpin to the connecting rod.

Aluminum Alloy Pistons.—Aluminum pistons outlined at Fig. 312, have replaced cast-iron members in most airplane engines, as these weigh about one-third as much as the cast-iron forms of the same size, while the reduction in the inertia forces has made it possible to increase the engine speed

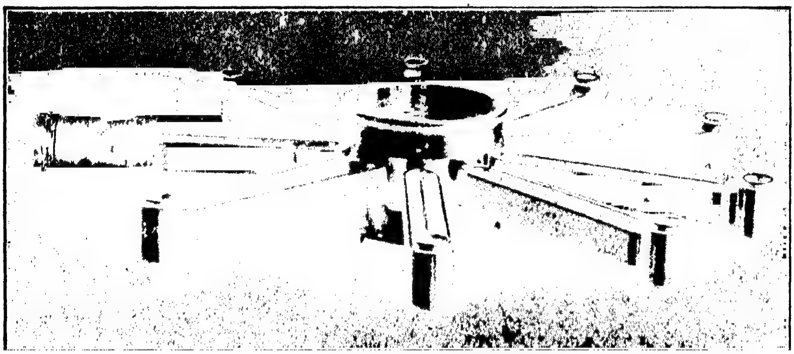


Fig. 311.—The Cast Iron Piston of an Early Gnome "Monosoupape" Engine, Installed on One of the Short or Link Rods.

without correspondingly stressing the connecting rods, crankshaft and engine bearings. Aluminum has not only been used for pistons, but a number of motors are built that use aluminum cylinder block castings as well as previously described. Of course, the aluminum alloy is too soft to be used as a bearing for the piston, and it will not withstand the hammering action of the valve. This makes the use of cast-iron or steel sleeves imperative in all aluminum block motors. When used in connection with an aluminum cylinder block the cast-iron pieces are sometimes placed in the mould so that they act as cylinder liners and valve seats, and the molten metal is poured around them when the cylinder is cast. It was said that this construction results in an intimate bond between the cast iron and the surrounding aluminum metal but experience has shown that this is not always true. Steel liners may also be pressed into the aluminum cylinders after these are bored out to receive them or screwed into suitably threaded bores as in the Hispano-Suiza engines. Aluminum has for a number of years been used in many motor parts. Alloys have been developed and described that have greater strength than cast iron and that are not so brittle. Its use for manifolds and engine crank and gear cases of all types has been general for a number of years.

At first thought it would seem as though aluminum would be entirely unsuited for use in those portions of internal-combustion engines exposed to the heat of the explosion, on account of the low melting point of that metal and its disadvantageous quality of suddenly "wilting" when a critical

point in the temperature is reached. Those who hesitated to use aluminum on account of this defect lost sight of the great heat conductivity of that metal, which is considerably more than that of cast iron. It was found in early experiments with aluminum pistons that this quality of quick radiation meant that aluminum pistons remained considerably cooler than cast iron ones in service, which was attested to by the reduced formation of carbon deposits thereon. The use of aluminum alloy makes possible a marked reduction in powerplant weight A small four-cylinder engine which was not particularly heavy even with cast-iron cylinders was found to weigh 100 pounds less when the cylinder block, pistons, and upper half of the crankcase had been made of aluminum instead of cast iron. Alumi-

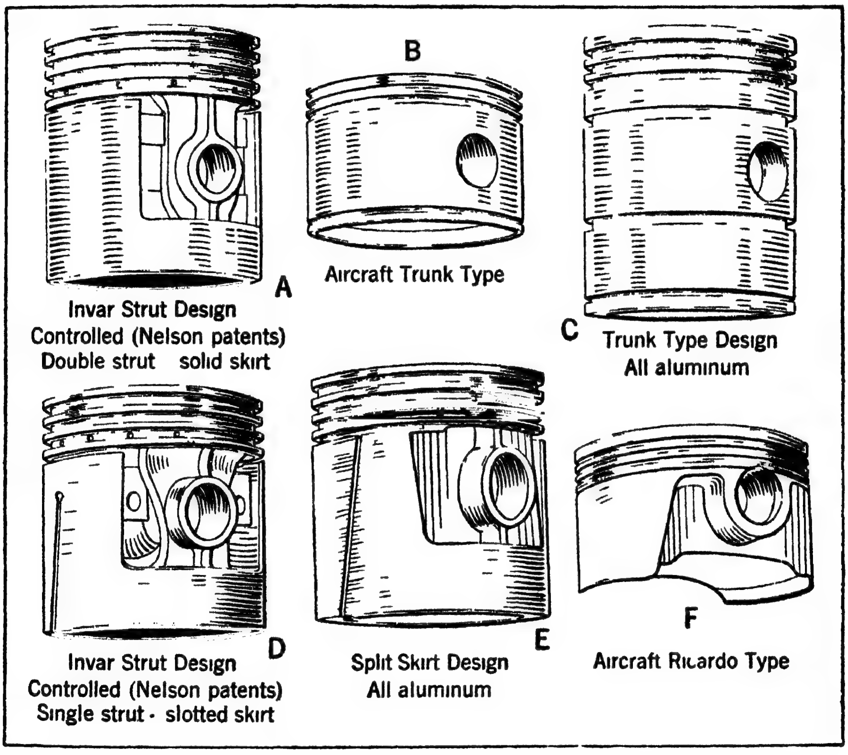


Fig. 312.—Some Typical Alloy Pistons Suited for Modern Automotive Engines. A—Double Invar Strut Design. B—Aircraft Trunk Type. C—Marine and Oil Engine Trunk Type. D—Single Strut, Slotted Skirt Design. E—Split Skirt Design. F—Ricardo Type.

num water-jacket block motors are no longer an experiment, as a considerable number of these have been in use in airplanes. Absolutely no complaint has been made in any case of the aluminum motor and it was demonstrated, in addition to the saving in weight, that the motors cost but little more to assemble and cooled much more efficiently than the cast-iron form. One of the drawbacks to the use of aluminum is its high price compared to iron for commercial use in large engines of automobiles and boats but this does not apply to aircraft motors.

Modern Pistons Well Developed.—Pistons have received a considerable amount of detail development work, but the general design has not undergone any great amount of change for a number of years. The most desirable characteristics in the piston are lightness, a low coefficient of expansion, good wearing qualities and good heat conductivity in the larger sizes. Lightness is attained by reducing the sections, if of cast iron, to a minimum. In small sizes the minimum thickness is as low as $\frac{5}{64}$ inch, and represents a fine example of the foundry art. Aluminum-alloy pistons are used successfully in practically all aviation engines and in most automobile motors and have the advantage in the matter of weight over cast-iron pistons, even though it is necessary to employ somewhat heavier sections. The fact that aluminum is a better conductor of heat than cast iron works to advantage in large-size cylinders, especially for aircraft work where, if cast-iron pistons were used, the center of the piston head would attain excessively high temperatures which would cause preignition. The fact that aluminum expands more than twice as much for a given increase of temperature as compared with cast iron necessitates the use of increased clearance between the piston and the cylinder when using aluminum pistons. The result is that, when the engine is cold, a considerable amount of piston slap may be encountered with aluminum pistons of the trunk type and there are also greater chances for compression leaks and oil seepage when the engine is cold past the piston with this greater clearance. These factors contributed to make the trunk type aluminum piston not altogether satisfactory for passenger-car use but improvements and change of form has made it extremely satisfactory for aircraft engines.

Owing to the good bearing qualities of aluminum it is customary to allow the aluminum to come into direct contact with the piston-pin, which is made a somewhat tight fit in the piston when cold, and the expansion of the piston permits the pin to creep when hot. The piston-pin is generally allowed also to work in the upper end of the connecting-rod, which is fitted with a bronze bushing. Other types of piston have been made but have not achieved any particular success. These include two-piece pistons, comprising a steel head and a cast-iron skirt or an aluminum head and steel or cast-iron skirt and one-piece pistons which are cut away to present bearing surface only on those areas where the thrust is distributed, or where it is necessary to support the rings.

Pistons are generally supplied with three piston rings located in individual grooves which are machined above the piston bosses. Sometimes an additional ring, called a scraper ring, is located near the bottom of the skirt in automobile engines so that its lower edge slightly overtravels the cylinder bore. This lower ring is sometimes effective in preventing an excess of oil from reaching the combustion-chamber and has been used with aluminum pistons of the trunk type to overcome the over-oiling tendency with the greater clearance. The most satisfactory piston rings are of the concentric type, in which the wall thickness is uniform all around the ring. The varying degree of elasticity which is required in the ring in order that it should tend to expand in a circle and fill the cylinder bore is attained by certain peening methods. The butting ends of the ring are generally finished to form a diagonal joint at 45 degrees. There has been much experi-

mental work in the matter of specially designed piston rings which attempt to form a more perfect seal than is afforded by the lapped joint. The fact is that the percentage of leakage past the joint is such a small part of the whole that any improvement in this regard yields practical results too small to measure. On the other hand, the simple ruggedness of the plain ring, as contrasted with the complicated construction associated with the majority of special piston ring designs, has everything to recommend it for airplane engine service.

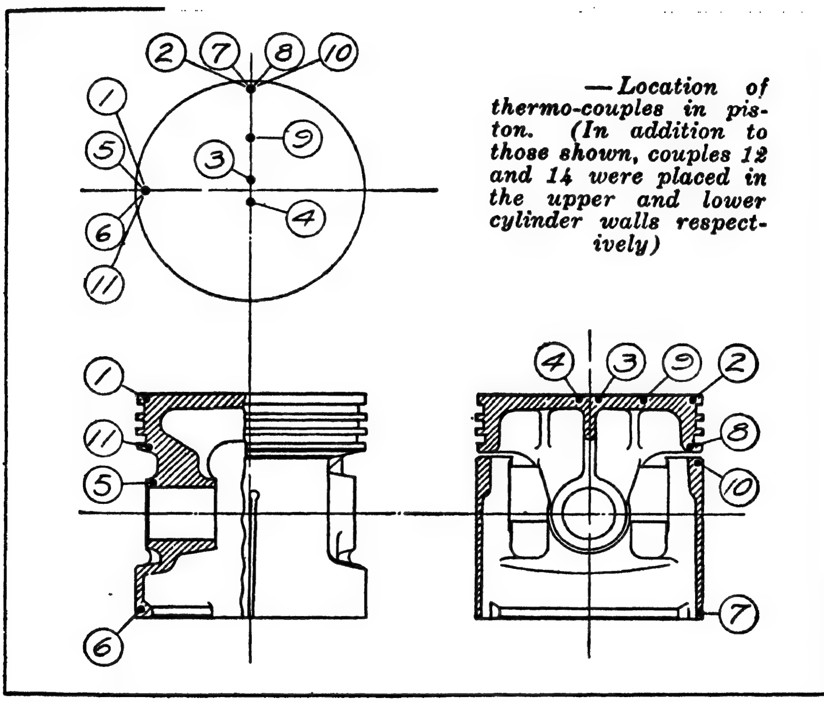


Fig. 312A.—Where Thermo-couples Were Placed in Piston to Determine Operating Temperatures at Different Points.

Some Problems in Piston Design.—Aviation engine piston design presents many major problems to the engine designer. High speed introduces inertia problems, high pressure requires strength, and the heat conditions call for thermal conductivity. The essential features in a successful piston may be listed as follows, without regard for their order of importance, the summary having been made by H. A. Huebotter, M.S.A.E., in the *S. A. E. Journal:*

(1) Light weight, to minimize inertia forces
(2) High thermal conductivity and emissivity, to improve heat dissipation; or refractory composition, to withstand high temperatures
(3) Proper bearing fit in the cylinder at all operating temperatures
(4) Strength adequate to withstand the maximum gas pressure in the cylinder
(5) Effective sealing against the gas in the combustion-chamber and against the lubricating oil in the crankcase

(6) Good bearing properties with normal lubrication
(7) Ease of fabrication
(8) Resistance to corrosion in ordinary service.

A glance at the above requirements suffices to show that a piston must be the product of a number of compromises. A piston designed for high speed alone will fail when applied to heavy service, as it has not enough metal to conduct the absorbed heat nor to carry the high fluid-pressures A piston that runs cool indefinitely at full loads is too massive for high speed. The less important quality must always abdicate in favor of those which are imperative.

Fortunately, a light alloy of aluminum provides material that combines low thermal-resistance and low weight. With a conductivity 3½ times that of gray iron and a density 40 per cent as great, such an alloy offers alluring possibilities to the piston designer, the more so when we consider that the conductivity of aluminum alloy improves as the piston becomes hotter, whereas the reverse is the case with gray iron. The automobile industry has consumed by far the greatest number of these practicable light-alloy pistons. However, other manufacturers of internal-combustion engines have played an aggressive part in applying aluminum-alloy pistons to their product. This development has taken place in response to the same trend in large-engine design that has marked automobile engine growth since its infancy. When high crankshaft-speeds and high mean-effective-pressures are adopted, the piston problem becomes acute and alloy pistons are drawn into service.

Among the present consumers of aluminum-alloy pistons in heavy-duty service are the motorboat and the gasoline rail-car manufacturers as well as aviation engine manufacturers. Popular engines for these services develop from 150 to 250 brake horsepower, with cylinders of 6- to 7½-inch bore. Such marine engines show 90 to 110 pounds per square inch brake m.e.p. at the peak of their torque curves. Considering the duration of service expected of these engines between overhauls, the operating conditions are rather severe, as is the case in aviation engines where brake mean effective pressures may run as high as 150 pounds per square inch.

Magnesium Pistons.—Magnesium has some advantages over aluminum, of the same nature as those that aluminum has over cast iron, but not to the same degree. It has no advantage in thermal conductivity, but it has a considerable advantage in specific gravity. A magnesium piston can be made from 30 to 40 per cent lighter than an aluminum piston. However, the process for reducing the metal is not nearly so old nor so highly perfected as is that for aluminum, and the price is consequently much higher. The same is true of the art of casting. Nevertheless, the engineering world, always looking for the optimum results, recognizes the possibilities of magnesium as a piston material and there has already been one instance where the metal has been tried out on a fairly large commercial scale. Considering that this was the first commercial installation, it gave a good account of itself, but it was discarded in favor of aluminum for much the same reasons that caused the early aluminum-piston installations to be discarded, because the price is high and the designs and technique necessary for large-scale production are not yet thoroughly worked out. Magnesium is ap-

parently more susceptible to corrosion by acids or alkalis than aluminum is so it is not as well suited for work in exposed positions or where chemical reactions may result. Its high cost and greater fragility also militates against its use but metallurgists are experimenting with alloys and the magnesium metal offers advantages that make it worthwhile when developed in a practical manner.

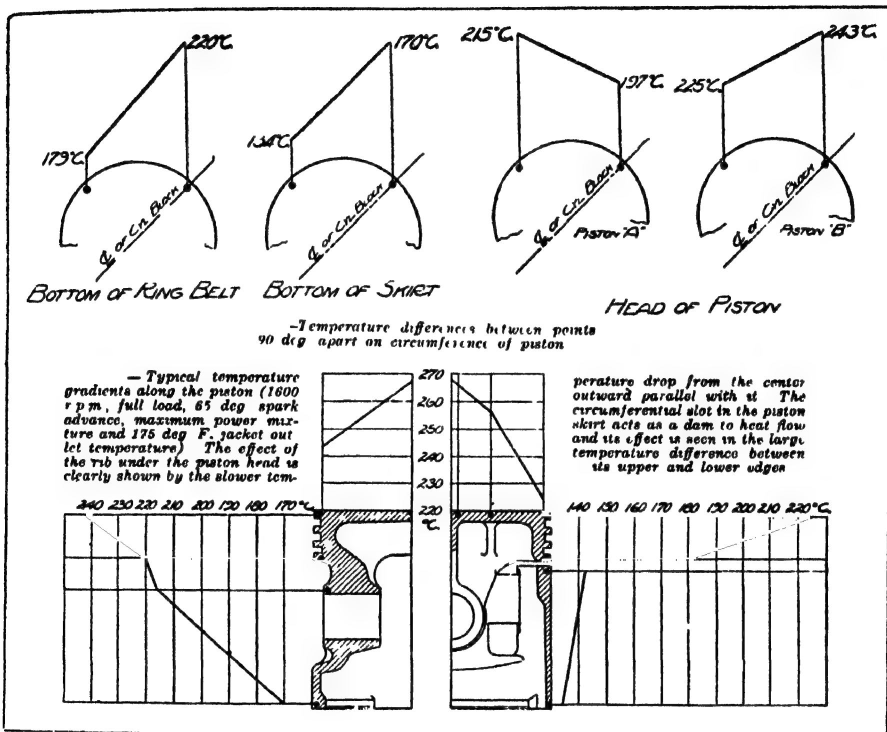


Fig. 312C.—Temperature Readings at Different Points on an Alloy Piston of the Split Skirt Type.

Operating Temperatures of Alloy Pistons.—In connection with an investigation of the factors influencing preignition, a study of the operating temperatures of aluminum-alloy pistons was recently undertaken in the dynamometer laboratory of the AC Sparkplug Company, Flint, Michigan, and described in *Automotive Industries.*

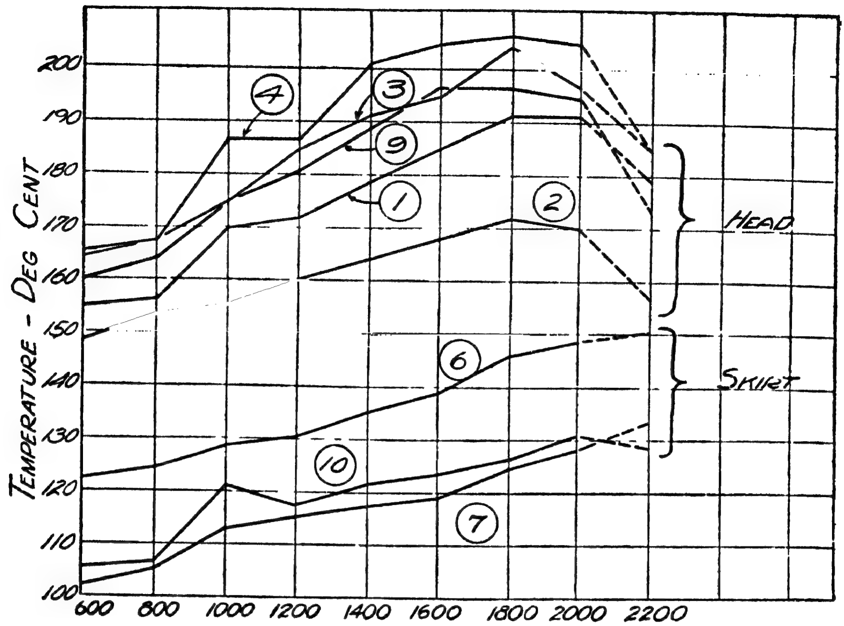
In making the experiments it was desired to find:

A. The maximum piston temperatures likely to be encountered in actual service with the particular type of engine and piston used in the investigation.

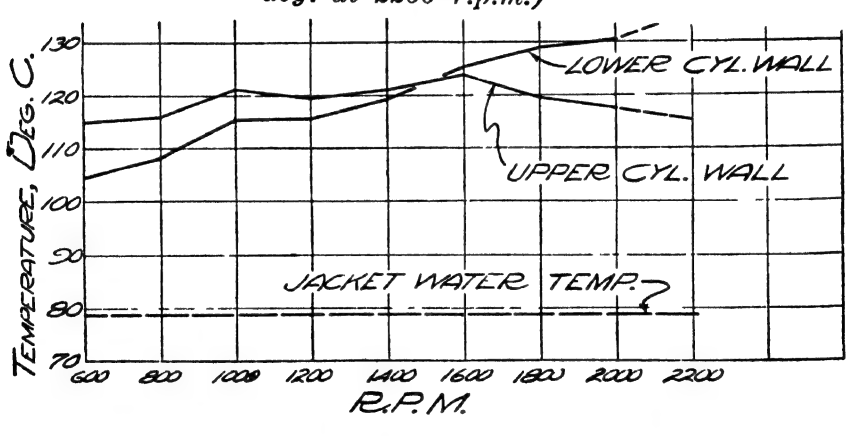
B. The effect of engine conditions upon these temperatures.

C. The temperature distribution throughout the piston, as far as practicable.

Because of the multiplicity of points at which the temperature was to be measured, the use of thermocouples was decided upon. As a check upon the accuracy of this method, a series of alloys of known melting points



—*Full-load runs. (Jacket outlet temperature 175 deg. F. Spark advance was inadequate for full power beyond 1600 r.p.m. and was retarded to 20 deg. at 2200 r.p.m.)*



—*Cylinder wall temperatures, showing influences of frictional heat and heat of combustion*

Fig. 312B.—Graphical Charts Showing Piston Wall and Cylinder Wall Temperatures Obtained by Tests.

were prepared, for use in a second piston at the most important positions The thermocouples and incidental apparatus operated successfully at the comparatively high speed of 2,200 r.p.m. A piston was fitted with eleven copper-constantan thermocouples in the positions shown in Fig. 312 A. No. 32 double cotton-covered wire was used. All couples were held in the

piston by peening or by staking from an adjacent hole and were installed from the inside of the piston, 1/16 inch below the outer surface, which was not disturbed by drilling through. Each thermocouple had a separate cold junction, a Leeds & Northrup potentiometer being used to measure the thermal e.m.f. Special linkage permitted the attachment of the wires to the reciprocating piston and suitable conduits attached to and moving with the piston permitted temperature determinations to be made while the piston was in normal service in the cylinder. The temperature readings obtained by this means are clearly shown in accompanying charts at Figs. 312 B and 312 C, which are self explanatory.

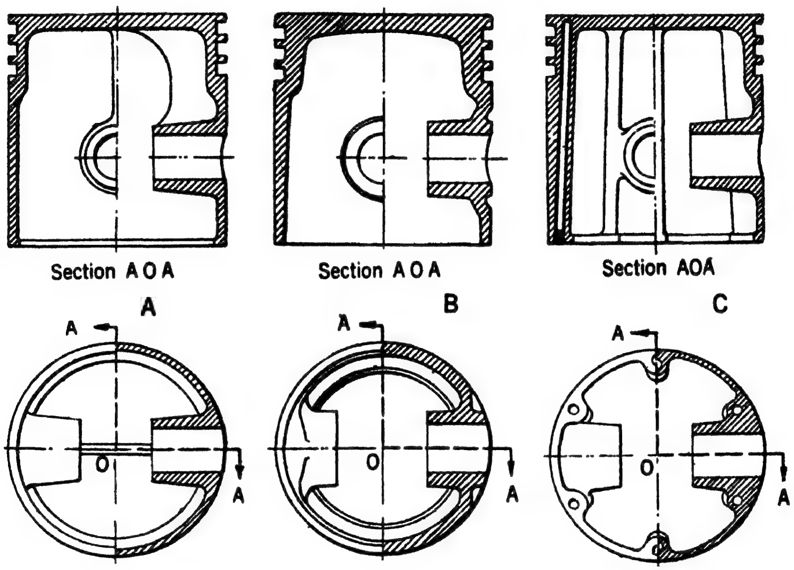


Fig. 313.—Sectional Views of Early Alloy Pistons Employed in Automotive Engines. A—Simple Trunk Type. B—Trunk Type Relieved at Wristpin Boss. C—Trunk Type with Drilled Heat Conducting Vertical Bosses Extending from Head to Skirt.

Aluminum Alloy Piston Forms.—There is such a wide difference of opinion concerning the fundamental theories that we may expect and do find a great difference in the design of pistons now used in internal-combustion engines according to E. G. Gunn, M.S.A.E., in a paper read before the S. A. E. and printed in the journal of the society. We find long pistons, short pistons, some with the piston-pin near the top, some with the piston-pin near the bottom, thick heads with cooling ribs and thin heads with no ribs.

From a thermal standpoint aluminum pistons may be broadly divided into two classes:

(1) Those designed with the object of conducting the heat away from the head into the skirt and thence into the cylinder walls

(2) Those designed with the object of partly insulating the skirt from the heat of the piston head.

Those in the first class are usually more or less conventional in design except that they have thicker walls, or ribs extending down from the head They are generally used for high-duty marine type engines. In these engines, which are usually of a rather large bore and comparatively low speed, the weight of the piston is secondary to its ability to keep the head from overheating, and piston slaps are not of much consequence. Therefore, for heavy-duty engines pistons of the first type seem logical. A sketch of some of the types which come under the first group are given in Fig. 313, A, B and C. The form at A has a heavy rib extending from the pin bosses to the head in the interior of the piston. That at B uses an extremely thick head. The form at C has a series of hollow vertical bosses running from the piston skirt to the head.

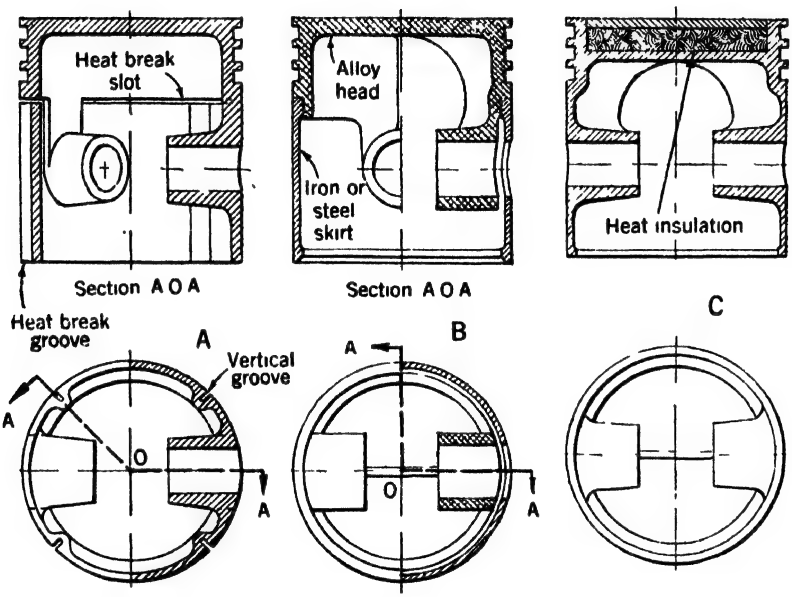


Fig. 314.—Sectional Views of Various Piston Designs Intended to Prevent Excessive Heat Transfer. A—Piston with Longitudinal and Vertical Grooves. B—Composite Piston Using Alloy Head with Iron or Steel Skirt. C—Piston with Heat Insulating Material in the Head, an Experimental Construction.

For passenger-car engines the conditions are somewhat different. The duty is lighter and the bore usually smaller. This lessens the tendency toward excessive heat. Quietness being important, close-fitting pistons are desirable. Need for good accelerating ability and smoothness in operation makes lightness desirable. These considerations have led to much development work on pistons of the second type. The plan followed in most cases is to partially insulate the skirt from the piston head by a heat gap, thus minimizing the expansion of the skirt due to heat. Some of the ways of accomplishing this are shown in Figs. 314 and 315. The methods shown at Fig. 314 are not widely used for various reasons. The piston at A not only has circumferential but vertical slots as heat breaks. That at B is a

composite construction with an alloy head screwed to a cast-iron or steel skirt. The form at C has a space in the head filled with insulating material but its value is doubtful.

Pistons of the Long and Franquist type are split to allow the piston to spring. They can for this reason be fitted more closely than the more conventional types. The conventional type with the comparatively thin wall is probably the most popular and for the smaller bores serves very well. It is simpler and somewhat cheaper to make than other types and is shown at Fig. 315 A. In all the other types shown there has been an attempt to insulate the skirt from the head. This allows the piston to be fitted more closely, thus minimizing piston slaps.

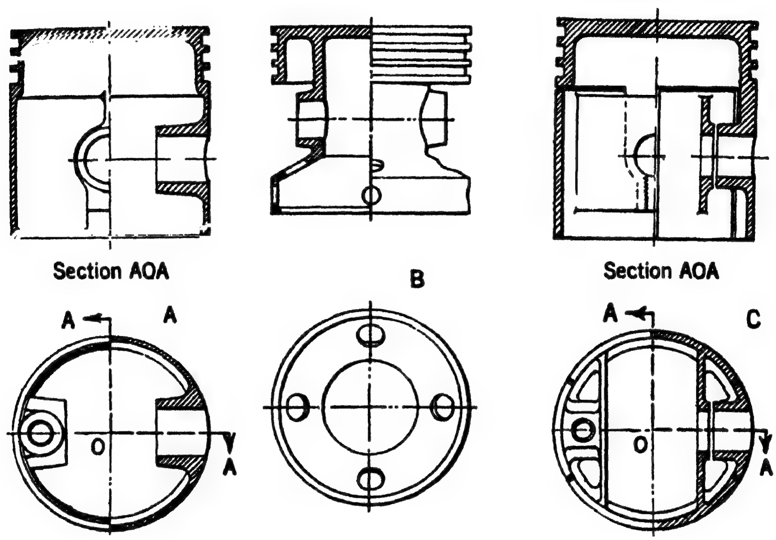


Fig. 315.—Alloy Pistons Suited for High-Speed Engines. A—Design of Piston with Very Light Walls. B—The Zenith Type Piston. C—Strut Type Piston with Heat Break Slots.

Four points are often brought up as objections to the use of aluminum pistons. These objections are the same as those encountered in the use of cast-iron pistons.

(1) Wear
(2) Piston slaps
(3) Excessive oil consumption
(4) Crankcase dilution.

Smooth Finish Important.—Wear has been shown to be largely a function of original smoothness. It is unreasonable to expect long life when aluminum pistons are fitted to cylinders of relatively hard material having a rough bore. Aluminum has been shown to be a good bearing metal but it is relatively soft and it must run on a smooth surface, as in the case of babbitt metal. Much attention is paid to polishing the journals of a crankshaft, but we often see cylinder bores, whether ground or reamed, which

are so rough that they can be marked with a lead pencil, although they may appear to be smooth. This is undoubtedly the cause of a great deal of initial wear on aluminum pistons. When cast-iron pistons are fitted this is not so apparent. Dust in the air also plays a very important part in wear. Automobile engines run on the dynamometer give much longer service than engines in cars which resulted in fitting air cleaners to all modern automobile engines but air cleaners are not necessary on aviation engines unless the airplane is used for student instruction and frequent landings and "take offs" on a sandy or dusty field are made. Under such circumstances they might prove advantageous.

Causes of Piston Slap.—Piston slaps can be overcome by using proper clearance. Pistons of the second design tend to make this condition easier to meet. Offsetting the piston-pin also tends to reduce piston slap. With 3½-inch and smaller bores there should be no trouble due to sticking with pistons of the conventional design fitted closely enough to prevent slaps, provided the piston and cylinder are of proper design. There should be no local hot-spots, and care should be taken in the design to prevent a condition tending to warp the cylinder when heated. Much trouble was experienced with sticking aluminum pistons in a certain inserted-sleeve engine of about 3½-inch bore. The cylinder was in the form of a block aluminum casting, with inserted cast-iron sleeves. Clearances up to $\frac{1}{64}$ inch were tried, but still the pistons seized. The sleeves were removed and found to be machined so that there was an air-gap of 0.005 inch between the sleeve and the cylinder wall. These sleeves were replaced with others which fitted all the way down, and pistons with 0.007-inch clearance were then found to be satisfactory. This is perhaps an exaggerated case, but shows the bad effect of failure to carry the heat away from the cylinder bore rapidly. The top land of the piston must, of course, be given much more clearance than any other part. The next land requires less, and the least clearance can be given to the bottom of the skirt. The tapering necessarily increases rapidly as the top of the piston is approached. When the piston-pin is placed too near the rings, piston slaps are more frequent, for the clearance in the zone near the piston-pin bosses must be sufficient to take care of extreme heat conditions so that under ordinary running conditions this part of the piston has enough clearance to allow piston slaps. When the pin is placed farther from the head the clearance can be small enough to prevent slaps.

Some trouble has also been encountered due to fitting pins tightly in the piston. When a piston with a tight pin is heated, it expands and creeps out on the pin; when it contracts again, it hangs to the pin so that the piston has a greater diameter parallel to the pin and a smaller diameter at right angles to it. This condition makes seizing easier and slaps more pronounced. This creeping can be demonstrated readily by applying a blow-torch flame to the head of a piston fitted with a tight pin.

Excessive Oil Consumption.—When too much oil is thrown into the cylinder bores, tight-fitting pistons and special rings will not completely overcome the trouble. A great many tests have been run which show this conclusively, demonstrating that:

(1) With no control on the oil being thrown into the cylinder, rings which seal the top and bottom edge of the groove reduced the oil

consumption

(2) When the oil is properly controlled, the oil consumption is very low even with rings having an up-and-down clearance of 0.004 inches

(3) With the oil controlled and with hot water circulated through the engine, the volume of liquid in the oil-pan increased, indicating dilution with fuel which passed the piston and rings. This was independent of the kind of rings used.

The engine under test was run with a device arranged to heat the mixture to a temperature of about 160 degrees Fahrenheit. This was accomplished in such a way that the maximum amount of heat was applied when idling. The effect upon fuel vaporization was observed through a glass window and was clearly evident. The result was to diminish the amount of fuel in the oil-pan when idling, and the viscosity was not seriously affected. Before the installation of the heating device a black deposit was found on sparkplugs taken from the cylinders of cars on road test even when the oil consumption was very low. After the installation, the sparkplugs remained clean under all conditions. It has been common practice for a number of years to put a quantity of kerosene in the crankcase oil, when running-in an engine to allow the bearing parts to seat more quickly. It is fair to expect that crankcase dilution has the same effect and that more rapid wear follows; hence the need for minimizing crankcase oil dilution with unburned fuel.

Aluminum Pistons Run Much Cooler.—Comparative studies of piston-head temperatures with various types of piston and rings were made under Mr. Grimes' direction when he was connected with the Franklin Automobile Company. The results of these are shown in the curves of Fig. 316. Note that the cast-iron piston runs 150 degrees hotter than the aluminum type. It is also interesting to note that pistons fitted with smooth-faced rings run cooler than those using rough-faced rings. Mr. Grimes attributed this to the greater friction of the latter type. The readers' attention is directed to the greatly increased cast-iron piston head temperature over that of the aluminum type.

A paper on Heat Flow in Internal-Combustion Engine Pistons was presented at a monthly meeting of the Indiana Section of the S. A. E. by Prof. G. A. Young, head of the School of Mechanical Engineering, and H. A. Huebotter, research associate of the Engineering Experiment Station of Purdue University. In this paper the heat flow in pistons is analyzed mathematically and the results thus arrived at are combined with the results of experiment. The following conclusions are arrived at:

1. The temperature in the piston head depends more upon the quantity of metal present than upon its distribution. A uniform section, however, has an advantage over both a tapered and a parabolic section.

2. The tapered barrel section is superior to the uniform section in its ability to disperse heat with minimum weight.

3. The ring belt disperses between 60 and 85 per cent of the total heat absorbed by the piston.

4. The ring belt transfers heat to the cylinder wall about 60 per cent as readily as the plain portion or skirt of the piston barrel.

5. In view of the low heat dissipating property of the ring belt, a broad band in contact with the cylinder at the head end of the barrel should be of material assistance in cooling the piston. By reason of this better cooling, it should not require the large clearance usually given to the top band in present designs, although the determination of the clearance is admittedly a delicate matter. Such a band would also help to support the piston and permit the use of a shorter barrel.

6. Improvement in the thermal contact between the rings and the piston barrel will assist in lowering the piston temperature.

7. In gray iron pistons a close grained iron in which the graphite is finely divided and uniformly distributed should be used, on account of its uniform high thermal conductivity.

Advantages of High Heat-Conductivity.—The advantages of high heat-conductivity are familiar to the engineer but are not so well known to the public. Chief among them are the possible employment of higher compression-ratios with less carbon deposited on the piston-head, resulting in less preignition, cleaner and cooler oil and cooler bearings. In addition to these things, the aluminum alloy possesses excellent machining properties, is hard and strong, and its wearing qualities are superior if it is properly lubricated. Aluminum has one property that is unfortunate for its use as piston material. That is a coefficient of thermal expansion more than twice that of cast iron. This property has been the greatest single obstacle in the way of the general adoption of aluminum pistons. If a plain cylindrical aluminum piston, similar to the conventional cast-iron piston, is fitted into the cylinder with the usual clearance allowed for cast iron it will expand so much that it will stick and score at high speeds or under heavy loads. If it is fitted with sufficient clearance to prevent this it will be so loose in the cylinder that it will rattle back and forth when the engine is started cold, producing the noise known as piston slap. To overcome this difficulty is the dream of every piston inventor. The vast majority of all of the over 1,000 light-alloy-piston patents deal with some scheme for compensating for the greater expansion of aluminum than of cast iron. This may be a peculiar method of slitting the piston, or it may be casting in or bolting on pieces of steel to act as expansion controls. It may propose rings of special types or a peculiar location of the piston-pin, but the object is almost always the same; to provide an aluminum piston that will not slap when cold and will not stick when hot. A few of these plans have been commercially successful and have played a major part in the rapid growth in the use of aluminum pistons.

Another factor that has played an important part in bringing the aluminum piston into prominence is the art of casting aluminum alloys in permanent iron moulds. This process was imported from France in 1912 in a crude state. The first piston castings by this process were produced in 1915. The growth of the resulting industry was slow for a few years, but as the technique advanced it became very rapid. Since 1920 many million aluminum pistons have been made by this process. Its advantages over sand casting are many. The metal is hard and fine grained, free from porosity and from blowholes, and has excellent machining properties. Great numbers of castings may be duplicated almost exactly as to both

dimensions and weight. Still another development in the last three or four years is the art of heat-treating aluminum pistons. By heat-treatment of permanent-mould pistons manufacturers are able greatly to relieve the casting strains which cause distortion in operation, and at the same time to increase the hardness to as much as 160 Brinell, thus enabling the piston to resist wear of the ring groove and skirt to a great extent.

Split Skirt Pistons.—The most familiar design of aluminum piston, the one that brought it into prominence, is the so-called slit-skirt design as shown at Fig. 312 E and Fig. 315 C. In this type of piston we have two bearing-faces of approximately 90 degrees each, separated from the head and ring section by horizontal slots. Up the center of one of these bearing faces, from the open end of the piston to the horizontal slot, runs a vertical slot. The purpose of the horizontal slots is to separate the bearing faces from the piston-head, preventing a direct flow of heat from the head to the skirt and thus maintaining a lower skirt-temperature. The purpose of the vertical slot is to allow the piston-wall to deflect slightly after it reaches the size of the cylinder bore, thus eliminating the possibility of seizure at high speeds. This is the design that has been in general use for nearly ten years in various automotive applications. The clearance provided with this type of piston is usually about the same as with cast iron.

Strut Type Piston.—The strut type shown at Fig. 312 A and D is identical with the split-skirt type in most of its essential features. It has the two 90-degree bearing-faces separated from the heat by horizontal slots. The piston-pin bosses are attached to the head exactly as in the split-skirt type. The connection from the piston-pin bosses to the edges of the bearing faces, however, instead of being by aluminum ribs, is by steel ribs cast into the piston-pin bosses and into the edges of the bearing faces, connecting the adjacent edges of the two bearing-faces in the same manner as do the aluminum ribs in the split-skirt piston. These two steel inserts carry the thrust loads of the piston against the cylinder-wall and act as spacers between the two bearing-faces, controlling the expansion across the bearing-faces due to temperature. If the inserts are made of cold-rolled steel the permissible clearance will be about the same as with a cast-iron piston of like diameter. If they are made of Invar, a steel containing about 35 per cent nickel, which has practically no expansion in the range of temperature encountered in piston operation, the clearance may be less than is ordinarily employed with cast iron. Pistons of this type are heavier and somewhat more costly to produce than the all-aluminum type so they have not been widely applied in aviation engines to date. They offer some problems in casting strain and distortion, incident to combining two materials of radically different coefficients of expansion in the same rigid structure, but they will operate with very small clearance and are practical in every respect.

Locking Wristpins in Alloy Pistons.—There are three practices of securing the piston pin in aviation engine pistons: (*a*) locking the pin in the piston and floating it in the rod, (*b*) locking the pin in the rod and floating it in the piston, and (*c*) floating the pin in both and using lock wires or brass buttons in the piston to prevent endwise motion of the pin. The design using the lock wires is the best where it is possible. The pin may

then be fitted quite tight in the piston by heating the piston in warm water and assembling with the rod and pin while the piston is warm. When the piston cools the pin will be quite tight in it and the oscillation upon starting the engine will be between the piston-pin and its bearing in the connecting-rod. After the piston is warm the bearing between the pin and the piston-pin boss will become free and the pin can then oscillate in the piston. This construction also makes possible a better design of the small end of the

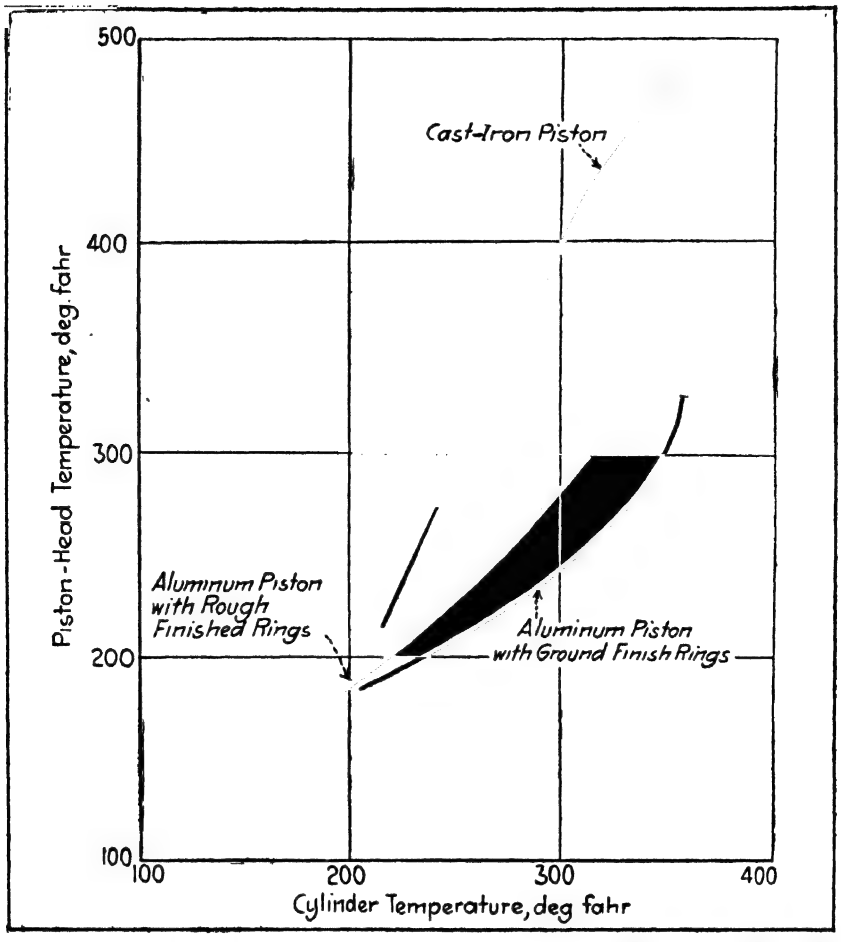


Fig. 316.—Comparative Piston Head Temperature Using Various Combinations of Rings and Pistons. Note that Pistons Using Rings Finished by Grinding Had Lowest Temperature.

connecting-rod, particularly for an aluminum connecting-rod, because it does away with the clamping device and is much stronger. Of the other two practices most engineers prefer locking the pin in the rod. In this case, the piston-pin should fit the piston with about 0.0005-inch clearance at room temperature.

Slipper Type Pistons.—In the ordinary form of piston, such as the trunk type, a large part of the piston surface is not needed to take the side thrust.

This unnecessary surface in contact is naturally prejudicial to mechanical efficiency. For this reason many pistons are relieved for part of their length by reducing the diameter about the piston-pin and are also drilled to reduce the surface in contact. The advantages obtained are real and distinctly shown on the test bench. Unfortunately, it has been found difficult to combine mechanical strength with lightness in these pistons and many failures have resulted from such attempts to side-track the defects of the piston of standard design. Further, the disposition of the bearing surface in pistons so relieved is not correct. There is still a 50 per cent excess of unnecessary surface, allowing for the fact that bearing surfaces have to be provided on each side of the piston, although only one side functions at a time. The rigidity of the piston-pin bosses is also seriously reduced unless the piston walls are unduly heavy. The orthodox trunk piston has been highly regarded by designers. It has simply been accepted without question. The success attending the various departures from conventional construction is a very useful object lesson that no established construction is beyond improvement.

One of the earliest of these departures is the Zephyr piston shown in Fig. 315 B. This type of piston has been very successful in aeronautic and racing engines, some of the first specimens being used on racing cars in 1912. In addition to the reduced bearing surface and adequate piston-pin boss support, it will be noticed that the design of the piston crown is well adapted to dissipating heat. A still further development toward a rational solution of the piston problem is that of the Ricardo slipper piston. This is shown in Fig. 312 F and Fig. 317. It will be seen that there are a considerable number of interesting points in this design, which have been tabulated in the *S. A. E. Journal* by Mr. Pomeroy as follows:

(1) The direct transmission of piston thrust to the slippers
(2) The proportioning of the slippers to the loads they carry, the compression slipper being reduced in area compared with that receiving the explosion thrust
(3) The slippers extend the whole working length of the piston and only laterally to the degree required
(4) Rigid support of the piston-pin bosses
(5) The ability to use a floating piston-pin
(6) Inherent lightness.

When a short type Ricardo slipper piston is used as in the form illustrated at Fig. 312 F it is not common practice to lighten the bearing shoes with holes as indicated in Fig. 317.

Slipper Pistons Increase Efficiency.—Mr. Pomeroy states that the improvement in power and reduction of gasoline consumption consequent upon the improved mechanical efficiency obtained with this type of piston varies from five to ten per cent at full load and is highest at high speeds. When it is remembered that an automobile engine is only developing about 25 per cent of its maximum power during a large percentage of running, and that under these conditions the inertia force and speed elements of piston friction are fully manifested, it will be seen that the effective increase in mechanical efficiency at low loads and high speeds may be very considerable. For example, about ten horsepower is required to propel a car

weighing 3,000 pounds at a speed of 30 m.p.h., with a gear ratio of 5.01; this corresponds to an engine speed of about 1,550 r.p.m. at which speed an engine of say 200 cubic inches capacity is easily capable of developing some 36 horsepower. The load factor, according to the conditions stated, is, therefore, only about 28 per cent. The well-known law of friction of machines, that friction is independent of the load, is as applicable to gasoline-engines running at constant speed as it is to the hand crane or pulley blocks

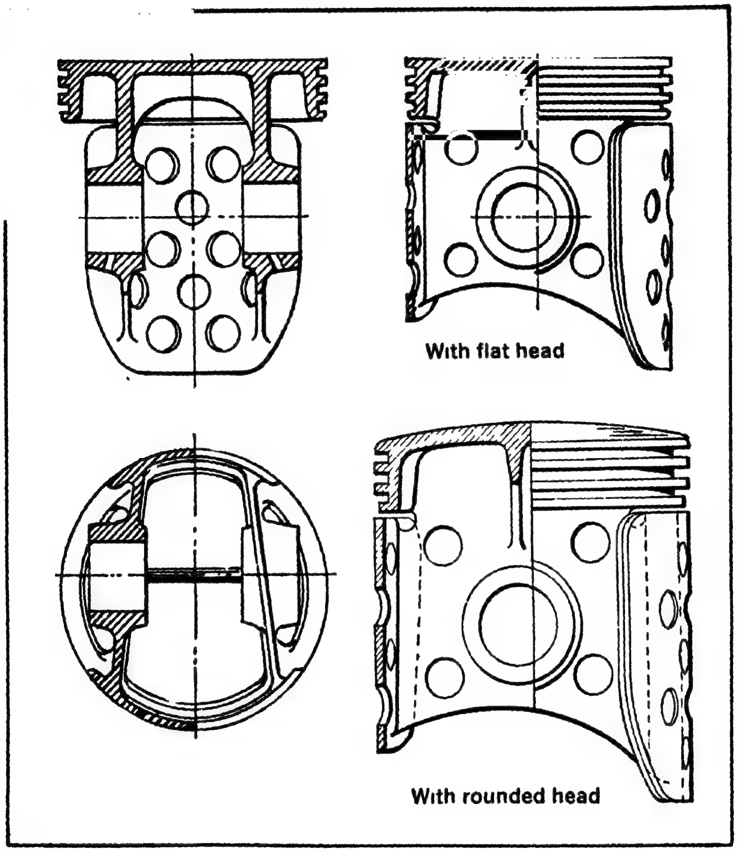


Fig. 317.—Sectional View Showing Construction of Flat and Round Head Types of Ricardo Slipper Piston Designed for High-Speed Engines.

of our school laboratories, especially as in the case under discussion the bulk of the loading is due to inertia effects which are constant by hypothesis. A very fair upper limit for the mechanical efficiency of a 200-cubic inch engine with cast-iron trunk pistons running at 1,500 r.p.m. is 87 per cent. The corresponding figure with slipper-type alloy pistons may easily be 92 per cent. The friction-horsepower for cast-iron pistons is, 5.2 horsepower, and for the slipper pistons=3.2 horsepower.

As cast-iron pistons are not used in aviation engines, the gain in efficiency by using Ricardo type pistons over the aircraft short trunk type shown at Fig. 312 B is not so marked but would be of sufficient value to warrant their use in engines of fairly large bore, providing the reduction in area permitted proper heat conductivity and adequate piston cooling.

Factors Affecting Clearance.—Different sections of a piston have different clearances, but the most important is the skirt clearance or that across the thrust faces. While many factors affect clearance, the more important include (1) the design of the piston; (2) the fitting of the piston from the mechanical point of view; (3) the machining, accuracy, and finish of the cylinder bores; and (4) the machining, accuracy, and finish of the pistons themselves. Other factors naturally include the thermal expansion and conductivity of both the aluminum alloy used and the cast-iron cylinder wall, variables such as peculiarities of the engine itself, the lubricating system, cooling system, and type of engine. On starting from cold, with a force-feed lubrication system, an appreciable time elapses before the oil is circulated. The piston clearance must therefore be adjusted to compensate for the increase in piston diameter due to initial heating. This situation does not arise, however, in engines using the splash system, oil being immediately thrown into the cylinder bores. The cooling system naturally has an effect on clearance, since it determines the engine operating temperature, and consequently that of the cylinder walls and pistons. Various factors affect engine cooling, including (1) capacity of the system, (2) area of radiator core, (3) type of system, and (4) amount of cooling air blast passing over fins of an air-cooled cylinder or through the interstices of a water-cooling radiator. The temperature of an engine is, of course, affected by the climate in which it is operated, whether a radiator shutter is used, and the nature of the cooling solution (water, water-alcohol mixtures, etc.). In some automobile engines using the thermo-syphon system, overheating is encountered on long trips and heavy pulls, and under certain operating conditions which are controllable by the driver. Hence special provision must be made as regards the design and clearance on aluminum-alloy pistons for these engines.

The coefficient of thermal expansion of an aluminum piston alloy is about 2.6 times that of cast-iron. However, the actual expansion of the head of an aluminum-alloy piston is not more than 50 per cent greater than that of a cast-iron head under the same engine conditions, because the head of the former does not get nearly so hot as that of the latter as shown by Mr. Grimes' chart at Fig. 316. Thus, in a calculated case, the actual expansion of the head of an aluminum-alloy piston was only 40 per cent greater than that of a cast-iron head. Actual experiments have shown that the cast-iron piston operates with a considerably higher head temperature than the aluminum piston, due to the lower heat conductivity of the cast-iron. Thus, reported figures indicate that the temperature of cast-iron pistons runs from 100 degrees Centigrade (212 degrees Fahrenheit) to 250 degrees Centigrade (482 degrees Fahrenheit) higher than that of aluminum-alloy pistons. The high thermal expansion of aluminum alloys is actually of little or no moment at the present time, since designs have been worked out

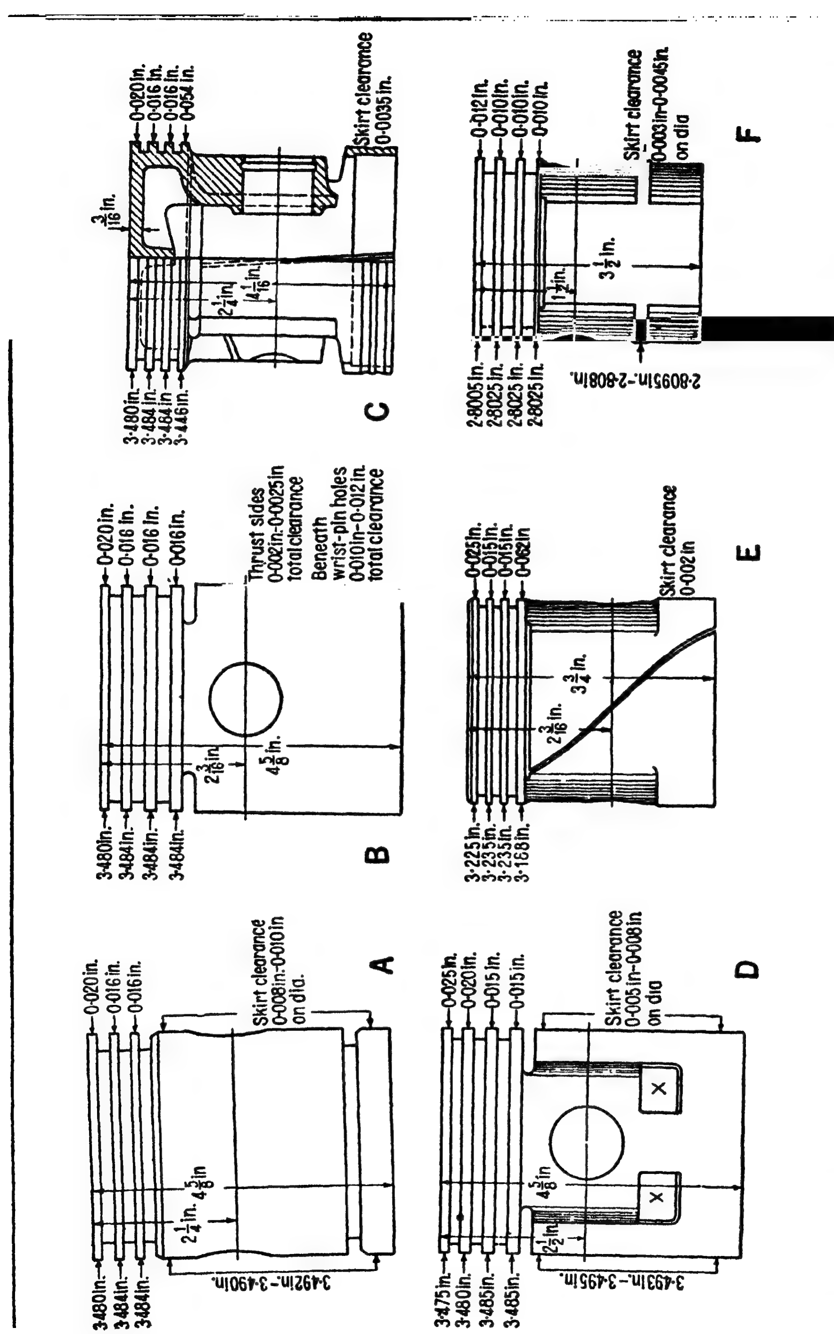


Fig. 318.—Diagrams Showing Recommended Clearances to be Allowed in Machining Various Types of Aluminum Alloy Pistons.

which adequately compensate for expansion and contraction due to temperature changes.

Cylinder Bore and Piston Finish Important.—In some engine manufacturing plants bores were formerly finished by rough boring and reaming. Recent advance in methods and precision of finishing cylinder bores has accompanied the development of the high-speed, high efficiency engine. Modern methods of finishing cylinder bores include (1) boring and grinding; (2) boring, grinding, and lapping; (3) boring, reaming, and honing; and (4) boring, reaming, and lapping, and various other combinations. The general tendency is toward final finishing by honing or lapping. In general, the larger manufacturers are producing cylinders by boring, followed by honing or lapping, although some plants finish by rolling, preceded by boring. For reconditioning engines, honing is making great strides, so much so in fact that the various regrinding associations are issuing propaganda designed to combat honing, lapping, and any method of reconditioning bores other than by grinding. While much may be said on this question, regrinding costs much more than honing or lapping which gives a desirable finish. Emery or lapping compounds are not to be recommended in finishing since the fine particles are forced into the pores of the cylinder and score both piston and cylinder wall, which practically eliminates the old expanding lead lap charged with abrasive. For cylinder bores, the usual allowance is 0.0005 inch in taper and out of round, and the preferred method of finishing is to grind or ream to this tolerance and then finish by honing or lapping to give a polished surface. A variation of 0.0005 inch in taper and out of round will, of course, affect the skirt clearance the same amount.

Assuming perfect machining, accuracy, alignment, and finish of both pistons and cylinder bores, the clearance required on aluminum pistons for a given engine depends upon the following factors: (1) The thermal expansion and conductivity of the alloys used for both piston and cylinder; (2) the design of the piston as regards sections, heat breaks, slots, etc.; (3) the engine operating temperature; and (4) the maximum and minimum temperatures attained by the piston head and cylinder block. As indicating the effect of improved design on clearance within the past ten years, the average skirt clearance for aluminum pistons has decreased from 0.003 inch per inch of piston diameter to 0.0005-0.001 inch (or, say, from 0.012 inch total clearance to 0.002-0.004 inch on a four inch piston). The older types requiring these greater clearances were solid-skirt pistons, in which no effort was made to control expansion. With the development in designs which permit close initial fitting clearance has come the development of better aluminum alloys having superior wearing qualities over those first brought out. Pistons are now manufactured to close specifications as regards hardness and strength. Further, the die-casting process, now used so extensively, gives a much superior casting.

At the present time, referring to the slotted-skirt piston, the average skirt clearance for aluminum pistons used in the usual water-cooled car engine is 0.0005-0.001 inch per inch of diameter; that for an air-cooled engine is 0.001-0.00125 inch per inch of diameter. The usual clearance for heavy-duty truck and 'bus water-cooled engines is 0.001-0.0015 inch, determined, of course, by the r.p.m. of the engine, load, and operating conditions.

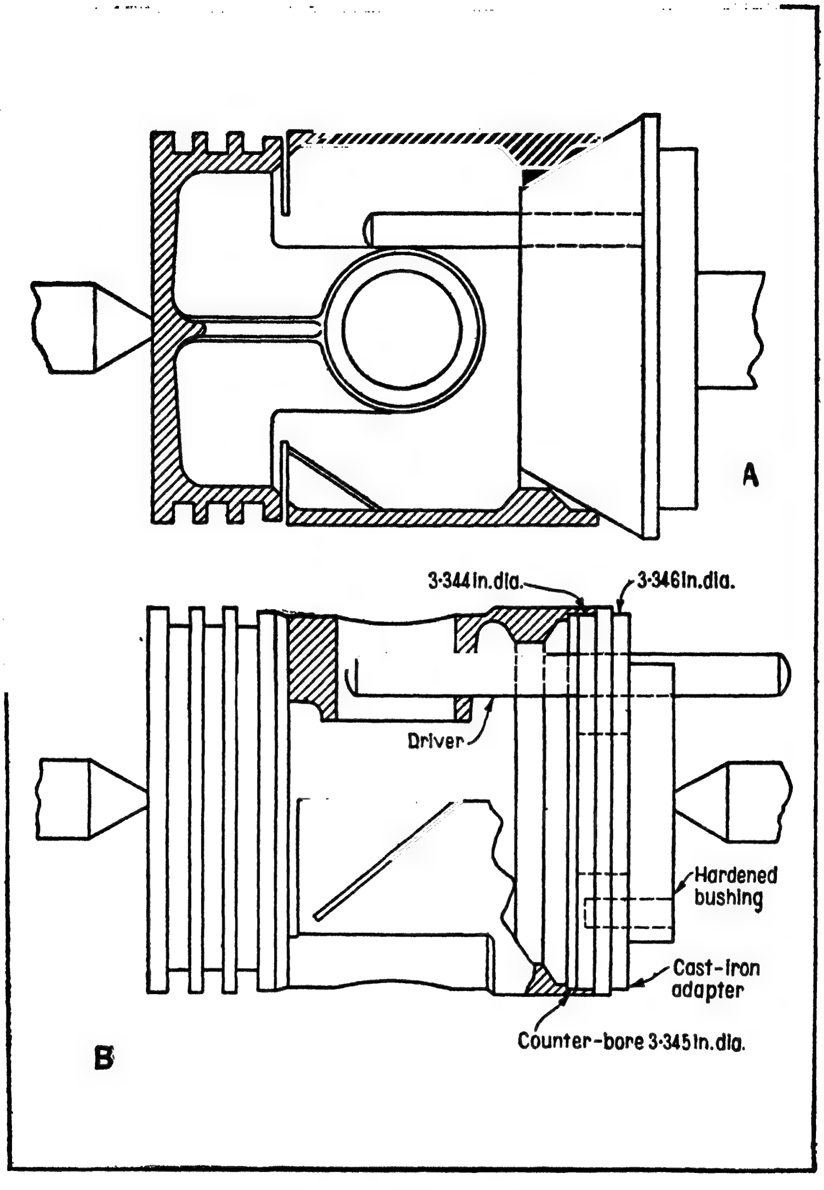


Fig. 319.—Methods of Supporting Alloy Piston for Machining. Cone Method at A is Not Adapted for Split Skirt Pistons on Account of its Expanding Effect. The Driving and Supporting Adapter Shown at B is that Best Suited.

The usually recommended skirt clearance for cast-iron pistons is 0.001 inch per inch of diameter or slightly less (e.g., 0.0008 inch). A number of aluminum pistons of various types and sizes are shown at Fig. 318. Pistons shown at A, B, C and D are intended to fit 3.5 inch bore cylinder. That at E a 3.25 inch bore while F shows the clearances on a still smaller piston.

Effect of Finishing Pistons.—The method of machining aluminum pistons and the grade of finish have a pronounced effect on the operating clearance. The usual procedure in volume production by engine manufacturers is to rough grind the piston casting to within 0.005-0.015 inch of the desired finished diameter, i.e., after turning, and then finish grind to size, removing 0.0005-0.0015 inch per traverse. When the piston is removed from the grinder, the slot, previously partially cut in, is completed at top and bottom. This method is advantageous as regards production, but is not the most desirable from the point of view of tolerance limits. When the slot is completed after finish grinding, the internal strains in the casting are released, causing a distortion to the amount of 0.001-0.008 inch on the diameter (of the skirt). Internal casting strains can, of course, be removed by heat treatment previous to finishing, but whether heat treatment be used or not, the slotting should be completed prior to finish grinding.

The best procedure in finishing pistons (and this is followed almost universally in the replacement field) is to rough turn to within 0.015 inch of the desired finished diameter, complete the straight, diagonal, or spiral slot at top and bottom, and then grind to the finished size. In this way, internal strains are relieved prior to the final grinding and subsequent distortion is avoided. Care must be taken that too much metal is not taken off per grinding traverse; excessive wheel pressure, will, of itself, cause distortion and possible fracture. In practice, also, it is to be recommended that the grinder be so adjusted as to finish the extreme skirt end of the piston 0.0005-0.0015 inch larger in diameter than the upper end of the skirt. This procedure will compensate for any drop due to the slight pressure of the step-cut adapter shown at Fig. 319 B which is fitted in the skirt end and holds the piston between centers for grinding. The exact amount of taper is determined by trial, and is considerably dependent upon the diameter of the piston. The grade and grit of the grinding wheel, wheel speed, and the grinding solution all affect the grade of finish. The grinding wheel recommended by experts for aluminum is the 36-L wheel manufactured by the Abrasive Wheel Co., Philadelphia, Pa.

The grinding fixture should exert no pressure which will cause distortion. In Fig. 319 A is shown a form of cone adapter. This has the effect of increasing the diameter of the skirt by spreading it, due to the pressure between centers. This will cause distortion, and the skirt will collapse when the piston is removed. Fig. 319 B shows a form of step-cut adapter developed by the Kant-Skore Piston Co. In this no pressure is exerted, and it is to be recommended in preference to either a cone or draw-bar fixture for use in supporting split-skirt type pistons for grinding.

Dycer-Austin Alloy Piston.—The Dycer-Austin aluminum-alloy piston shown at Fig. 320 has the same weight as the stock piston it replaces. There are four one-eighth inch ring grooves; three compression and one oil wiper ring. The design of new production aircraft engines takes cognizance of the fact that a narrow ring is more efficient, supplying greater compression, less friction and drag and less cylinder wear. A narrow ring is particularly advantageous for use in worn cylinders as reasonable efficiency is gained even in the presence of off-center operation. In regard to oil consumption, a 25 per cent increase in economy is realized due to closer fit of piston and

the use of narrow rings. Pistons are made with one-eighth inch higher head supplying greater horsepower and more revolutions to the OX5. Dycer-Austin pistons are also supplied with standard head when desired. Expansion of the detached head spreads the bosses apart and contracts skirt thus rendering the piston round under operating temperature. The piston skirt is turned oval .003 with the smaller diameter in the same axis as

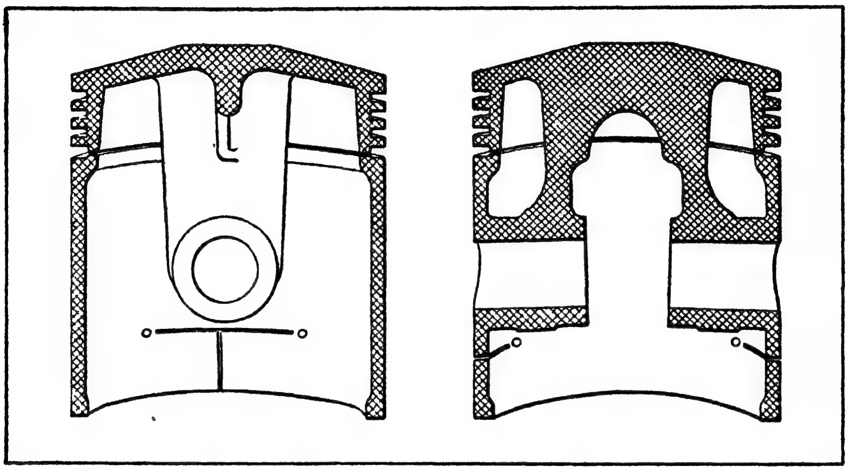


Fig. 320.—Sectional View Showing Construction of Dycer-Austin Replacement Piston.

the piston pin. As demonstrated by three years of service, there is no danger of the piston warping. Due to the connecting strut the detached head adds sixteen square inches of heat radiating surface to the piston thereby permitting a cooler operating temperature to the piston, rings and cylinder walls. The cut at bottom of skirt absorbs expansion, consequently preventing seizing in event of overheating due to lack of oil, etc. The slot serves to drain oil and prevent the travel of heat from the head to the skirt. Dycer Airport has used the special Dycer-Austin pistons in its own engines for some time. They have been in production only a short time. The high compression feature is interesting to the commercial plane owner today due to the increased efficiency realized with small cost. Incidentally, the motor requires overhauling at less frequent intervals due to its increased revolutions after overhaul and the extension of time before arriving at the usual overhaul period.

Durator Iron Piston.—A number of features are incorporated in the Durator piston which has been introduced recently by the Duroseal Corp., Cleveland, Ohio. Pressure at the piston head is applied directly to the pin bosses through rib structures while these bosses are entirely separate from a new type of skeleton skirt. In the design of the skirt, ample bearing surface is obtained and distortion due to the load on the piston head is said to be minimized. Perhaps the outstanding feature of the piston is the ring design. Provisions are made for but two rings, the lower of which is the slotted scraper type set in a drilled groove. The upper ring assembly con-

sists of three segmental units of low friction nonscoring alloy. Due to the segmental construction, this ring assembly exerts no pressure on the cylinder wall except during the compression and firing strokes and it is stated that the pressure when exerted is in approximate proportion to the pressure within the combustion-chamber, which tends to expand the segmental assembly to produce the sealing action. The action of this ring is somewhat like that obtained with the obdurator rings fitted on Gnome engines.

Fig. 321 illustrates the components of the piston and ring assembly. A flat head and short cylindrical section in which the rings are mounted

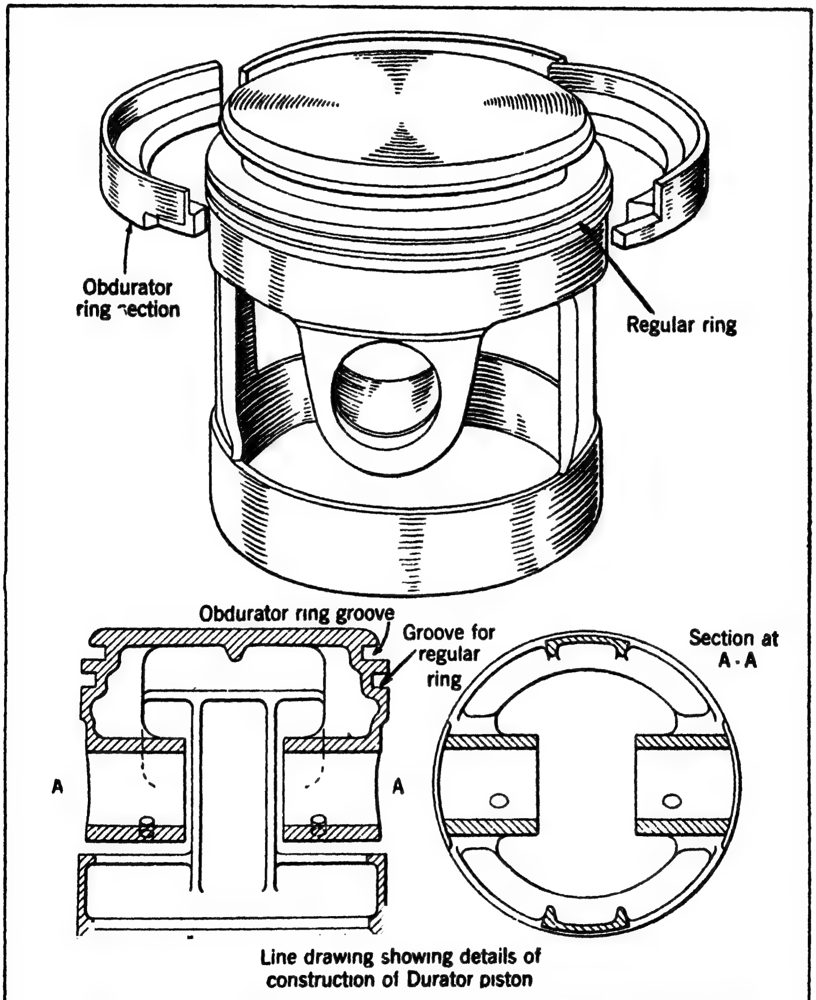


Fig. 321.—The Durator Piston is a Nickel-Iron Skeleton Member, Carrying One Scraper Ring of Conventional Form and a Special Three Piece Alloy Ring at the Top, which is Expanded by Gas Pressure. The Piston Bosses are Joined to the Heads by Two Ribs Each.

are joined to a ring forming the bottom of the skirt by two relatively narrow straps. As the piston is assembled in the engine, these straps are on the lateral center line and their outer surfaces are relieved so that the piston bears on the cylinder wall at the full ring at the bottom and a similar area located just below the scraper ring. The piston pin bosses are located between these two rings and are connected with the head by two ribs at each side which extend upward inside of the top of the piston.

Another feature which contributes to mechanical strength and elimination of oiling troubles is the shelf-like rib which extends around the inside of the piston at a point just below the drain holes for the scraper ring. This rib serves as a deflector for oil thrown off at the crankshaft and causes eddies which tend to draw off excess oil from the drain hole outlets. The ring forming the lower end of the piston is thickened slightly at the extreme bottom for additional strength and to provide a pilot for machining. This unit is cast iron of high nickel content which produces a hard tough alloy. Due to the skeleton construction, the weight is considerably less that that of the conventional design of cast-iron piston.

While the lower or scraper ring is set in a drilled groove of usual design, the upper ring assembly and its groove are unique. The groove is much deeper than ordinary and the top land of the piston is smaller in diameter and beveled off at the top. This arrangement provides room for the segmental ring assembly of L-shaped section. Obviously this assembly is inert and does not load the cylinder wall until pressure accumulating above the piston head tends to expand the segments. Sealing is assisted by the step joints in the segmental assembly. As operation is secured by gas pressure, no spring action is required and this assembly is made of nonferrous, nonscoring alloys rather than the usual cast iron. It is claimed that the new piston and ring assembly as installed in an engine tends to reduce friction and therefore increases the net output of the engine. As only one ring of the conventional type is carried by each piston, wall friction is at a minimum until pressure builds up over the head. It is claimed that the relative absence of wall friction tends to reduce vibration. With the reduction of friction loss, fuel economy is improved while the sealing action of the special top ring reduces dilution. Tests have indicated that the piston boss construction practically eliminates distortion of the skirt.

Piston Ring Construction.—As all pistons must be free to move up and down in the cylinder with minimum friction, they must be less in diameter than the bore of the cylinder. The amount of freedom or clearance provided varies with the construction of the engine and the material the piston is made of, as well as its size, but it is usual to provide more space at the head end than is left at the skirt to compensate for the expansion of the piston due to heat and also to leave sufficient clearance for the introduction of lubricant between the working surfaces. Obviously, if the piston were not provided with packing rings, this amount of clearance would enable a portion of the gases evolved when the charge is exploded to escape by it into the engine crankcase. The packing members or piston rings, as they are called, are split rings of cast iron, which are sprung into suitable grooves machined in the exterior of the piston, three or four of these being the usual number supplied. These must have sufficient elasticity so that they bear

tightly against the cylinder wall and thus make a gas-tight joint. Owing to the limited amount of surface in contact with the cylinder wall and the elasticity of the split rings the amount of friction resulting from the contact of properly fitted rings and the cylinder is not of enough moment to cause any damage and the piston is free to slide up and down in the cylinder bore, yet maintain a reasonably gas-tight joint. Various piston clearances are listed in the clearance tables of the various engine types to be described.

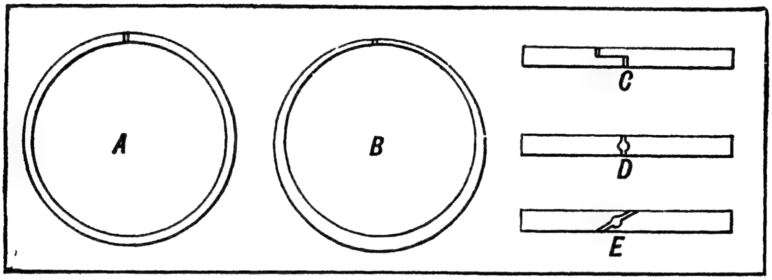


Fig. 322.—Common Types of Piston Rings and Ring Joints. A—Concentric Ring. B—Eccentrically Machined Form. C—Lap Joint Ring. D—Butt Joint, Now Seldom Used. E—Diagonal Cut Member is a Popular Form.

Concentric vs. Eccentric Rings.—These rings are made in two forms, as outlined at Fig. 322. The design shown at A is termed a "concentric ring," because the inner circle is concentric with the outer one and the ring is of uniform thickness at all points. The ring shown at B is called an "eccentric ring," and it is thicker at one part than at others. It has theoretical advantages in that it will make a tighter joint than the other form, as it is claimed its expansion due to heat is more uniform. The piston rings must be split in order that they may be sprung in place in the grooves, and also to insure that they will have sufficient elasticity to take the form

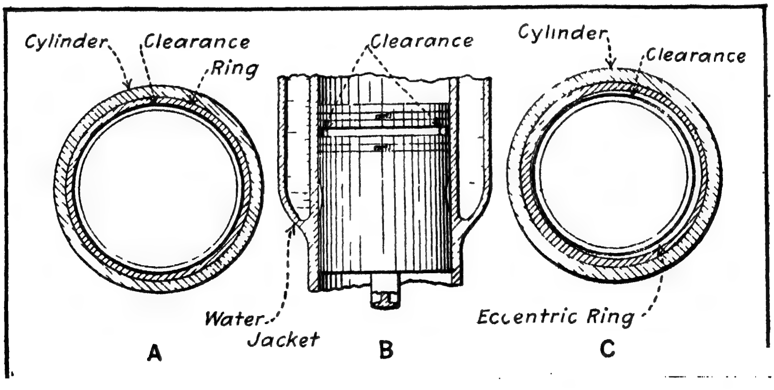


Fig. 323.—Diagrams Showing Advantages of Concentric Piston Rings. A—Note Uniform Clearance Between Back of Ring and Piston Groove with Concentric Ring. B—Sectional Diagram Showing Location of Clearances Between Inside of Ring and Bottom of Piston Grooves. C—Lack of Uniformity in Clearance Space When Eccentric Rings are Used.

of the cylinder at the different points in their travel. If the cylinder bore varies by small amounts the rings will spring out at the points where the bore is larger than standard, and spring in at those portions where it is smaller than standard.

It is important that the joint should be as nearly gas-tight as possible, because if it were not a portion of the gases would escape through the slots in the piston rings. The joint shown at C is termed a "lap joint," because the ends of the ring are cut in such a manner that they overlap. This is the approved joint. The butt joint shown at D is seldom used and is a very poor form, the only advantage being its cheapness. The diagonal cut shown at E is a compromise between the lap shown at C and the poor joint depicted at D. It is also widely used, though most constructors prefer the lap joint, because it does not permit the leakage of gas as much as the other two types. There seems to be some difference of opinion relative to the best piston ring type—some favoring the eccentric pattern, others the concentric form.

The concentric ring has advantages from the lubricating engineer's point of view; as stated by the Platt & Washburn Company in their textbook on engine lubrication, the smaller clearance behind the ring possible with the ring of uniform section is advantageous.

Fig. 323 A shows a concentric piston ring in its groove. Since the ring itself is concentric with the groove, very small clearance between the back of the ring and the bottom of its groove may be allowed. Small clearance leaves less space for the accumulation of oil and carbon deposits. The gasket effect of this ring is uniform throughout the entire length of its edges, which is its marked advantage over the eccentric ring. This type of piston ring rarely burns fast in its groove. There are a large number of different concentric rings manufactured of different designs and of different efficiency.

Figs. 323 B and 323 C show eccentric rings assembled in the ring groove. It will be noted that there is a large space between the thin ends of this ring and the bottom of the groove. This empty space fills up with oil which in the case of the upper ring frequently is carbonized, restricting the action of the ring and nullifying its usefulness. The edges of the thin ends are not sufficiently wide to prevent rapid escape of gases past them. In a practical way this leakage means loss of compression and noticeable drop in power. When new and properly fitted, very little difference can be noted between the tightness of eccentric and concentric rings. Nevertheless, after several months' use, a more rapid leakage will always occur past the eccentric than past the concentric. If continuous trouble with the carbonization of cylinders, smoking and sooting of sparkplugs is experienced, it is a sure indication that mechanical defects exist in the engine, assuming of course, that a suitable oil has been used. Such trouble can be greatly lessened, if not entirely eliminated, by the application of concentric rings (lap joint), of any good make, properly fitted into the grooves of the piston. Too much emphasis cannot be put upon this point. If the oil used in the engine is of the correct viscosity, and serious carbon deposit, smoking, etc., still result, the only certain remedy then is to have the cylinders rebored and fitted with properly designed, oversized pistons and piston rings.

Gray Iron Best for Rings.—Practically all piston rings in use at the present time are made of gray cast iron, and one little realizes what fine qualities this rather common material possesses until one reads the literature of the piston ring makers. What makes cast iron particularly suitable for this purpose is that it combines a fair degree of elasticity with a rather soft texture. It will therefore give the required spring pressure without unduly wearing the cylinder wall. Whatever wear takes place on the contact surface should preferably be on the rings, because the rings are much

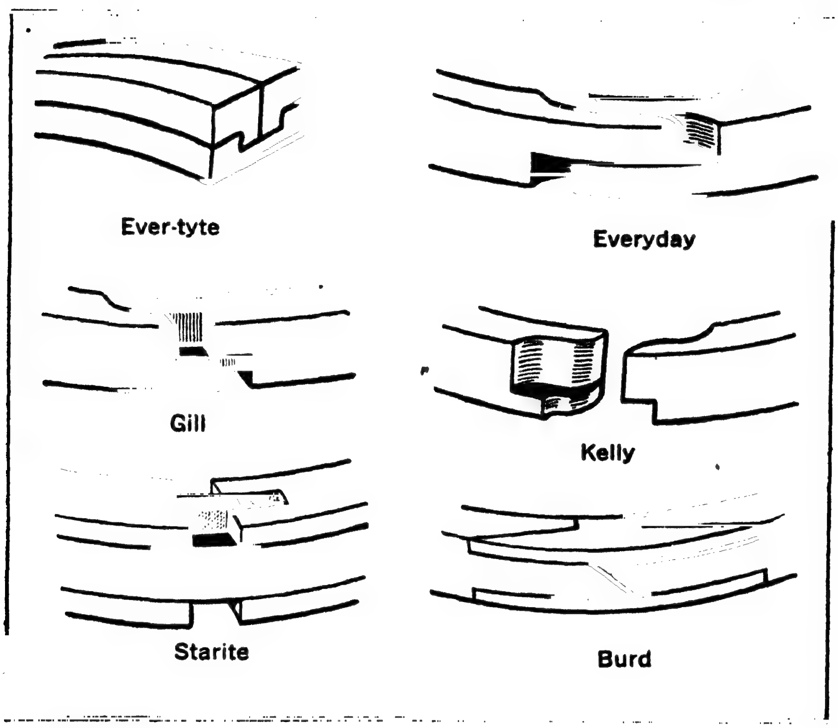


Fig. 324.—Special Ring Joints Intended to Provide Against Leakage at the Gap. Rings of this Type Have Been Used in Automobile Engines Successfully.

easier to replace than the cylinder. Other materials besides cast iron are being used to a slight extent, however. Thus, one concern has been making piston rings of monel metal, a natural alloy of copper and nickel; another makes one member of a two-piece ring of bronze; and several use steel springs under cast iron or bronze rings. Perhaps the best known use of bronze in aviation engines was the obdurator ring of the early Gnome engines.

While bronze, Swedish iron and even malleable iron and steel have been tried, it will be conceded that, so far, cast iron is the only satisfactory metal suitable for piston-ring usage in the internal-combustion engine. The density, the resiliency and the small cross-sectional area each being an important factor, it is evident at once that the foundry offers the greatest opportunity for improvement toward piston-ring perfection. Manifestly, with

poor castings at the start, very little better than poor results can be expected at the finish. There is little question as to the superiority of the individually cast over the the pot-cast piston-ring. Table below gives a mixture formula for individually cast rings as published in the *S. A. E. Journal.*

Extreme care in the selection of materials, combined with frequent physical tests, will be necessary to maintain the standard. A required property of a test-bar one-half inch square is a Shore hardness of 35 to 40 or a Brinell hardness of 200 to 230.

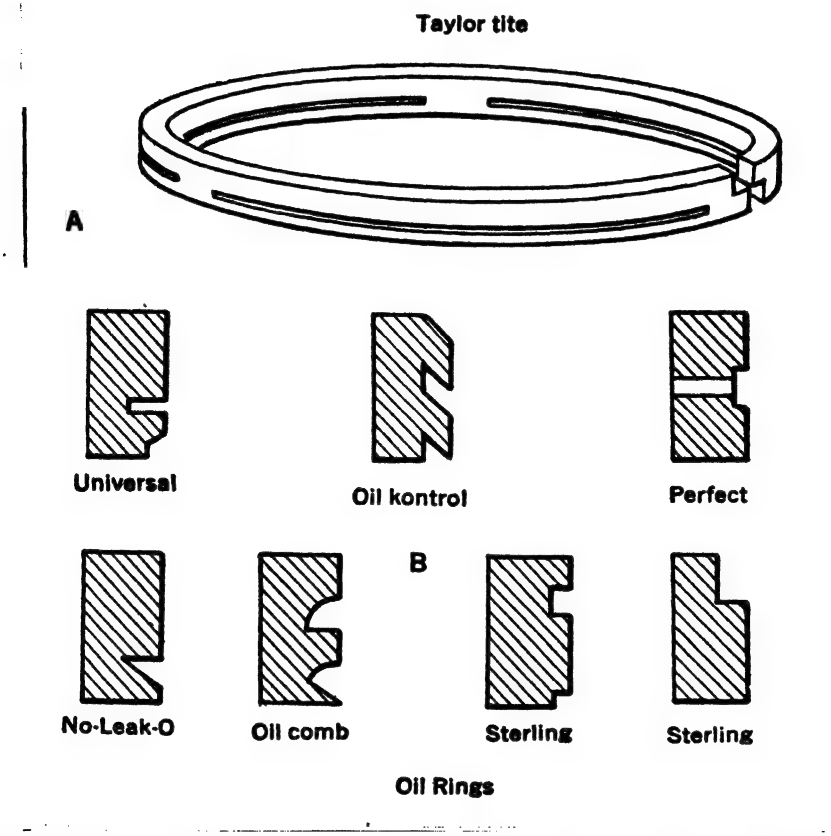


Fig. 325.—Ring With Oil Control Slots Shown at A. Various Systems of Machining Oil Control Grooves in the Face of Rings Shown at B.

FORMULA FOR INDIVIDUALLY CAST PISTON-RINGS

Substance	Per Cent
Silicon	2.50 to 3.00
Sulphur, maximum	..... 0.70
Phosphorus	0.30 to 0.50
Manganese	0.45 to 0.70
Combined Carbon	0.50 to 0.60
Graphitic Carbon	2.75 to 2.65

Remainder to complete 100% is iron

Rings Made from Individual Castings.—Originally piston rings were made from pot castings, and this method is still being followed by some makers, but the majority now make them from individually cast blanks. Where the individually cast blank is used the inside of the ring is left in the rough, or if it is worked upon practically only the burrs at the edges and any slight unevennesses of the surfaces are taken off by snagging. The surface metal of the ring is chilled in casting and therefore is of a closer grain, somewhat harder and more springy than the metal below the surface, and for this reason a ring with an inner unfinished surface, with the scale in place, is believed to be a better ring. Those who still make their

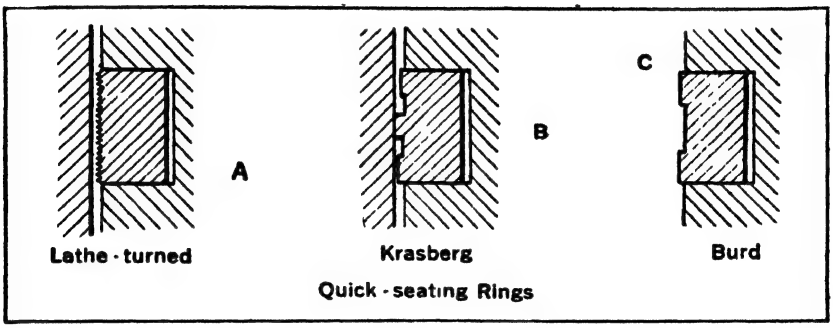


Fig. 326.—Diagram Showing Method of Machining Surface of Piston Rings to Insure Quick Seating of New Rings.

rings from pot castings sometimes claim greater uniformity of material in the ring, and this claim is evidently well founded, but the question then arises whether it is better to have the ring consist entirely of a metal of a certain quality or to have two kinds of metal in the ring, one the same as that in the ring of uniform material and the other a material of a higher quality. The advantage of the pot casting process would seem to be lower cost of manufacture, but the process of making rings from individual castings has been so developed that it is little if any more expensive to produce rings by it.

Reason for Peening Ring Interior.—There are various methods of endowing a ring blank turned or ground to the diameter of the bore, with the expansive qualities necessary to produce the desired pressure against the cylinder wall. The most commonly used consists in hammering or peening the ring on the inside, or on the sides near the inside edges. This peening process has the effect of spreading the metal where the blows fall, and the final effect is to increase the radius of that portion of the ring where the peening has been done. If the ring were peened uniformly over its whole inner circumference the radius would be increased uniformly all around, and, as with uniform pressure of the cylinder on the ring all around, the radius is reduced much more at the center than at the ends, this would not give a ring producing a uniform pressure. To get uniform pressure the peening must be intensified from the ends toward the center of the ring,

which can be done either by varying the intensity of the blows or by varying their spacing.

Another plan of putting rings under tension described by P. M. Heldt consists in expanding the ring, after it has been turned and gapped, over a taper mandrel or a mandrel of somewhat larger diameter than the inside diameter of the ring, and while in this expanded condition, subjecting it to a heat treatment. The effect of the heat treatment is to cause the molecules of the iron to set in the positions corresponding to the expanded condition of the ring, so that it requires pressure against the outside surface of the ring to bring it to its original position.

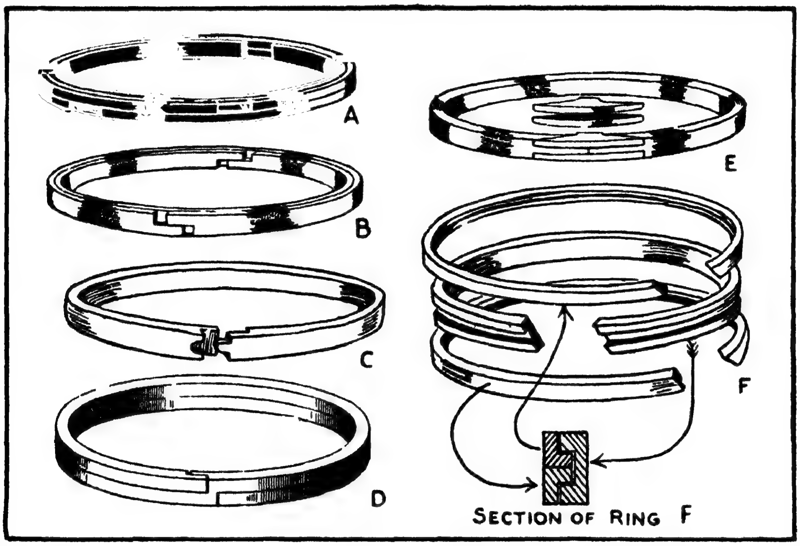


Fig. 327.—Illustration Showing Construction of Various Forms of Composite Piston Rings.

Piston Ring Width.—The proper width of piston rings is another subject productive of much discussion. The reduction in weight of all reciprocating parts is certainly to be desired. When it is taken into consideration that any added weight on a piston ring must be multiplied by a factor of six to determine its equivalent inertia effect for each cylinder, the advisability of narrow widths is evident immediately. Theoretically a knife-edge in contact with the cylinder wall will produce the proper result. It remains only to establish the added width necessary to allow for practical production, always considering that the minimum width is desired. Present methods of manufacture indicate that one-eighth inch is the proper width for aviation engines, all things considered. The thickness of the ring or the depth of the groove is subject to the same consideration in all ways. The merits of both eccentric and concentric forms have been discussed previously. No doubt, the eccentric ring is more correct for theoretical uniform wall-pressure. However, if the pattern for the casting is designed for the ring at its full opening, and the natural surface density of the inside of the ring is left undisturbed in machining, a proper foundry mixture will

produce a concentric ring with a wall pressure that is so nearly uniform in actual operation that its many other advantages make it preferable. It should be remembered also that the theoretical eccentric ring tapers down from its heaviest section, opposite the joint, to a knife-edge at the joint. Any design for an eccentric ring must modify this form to a certain extent, to avoid the easily breakable thin wall at the joint. To perform its function, the piston ring must exert a pressure against the cylinder wall, and this pressure should preferably be uniformly distributed over the whole of the contact surface of the ring. Unfortunately, there seems to exist no method for measuring the specific pressure exerted by the ring at any particular point of its contact area, and in default of such a method makers and users have resorted to methods intended to give the total pressure exerted by the ring.

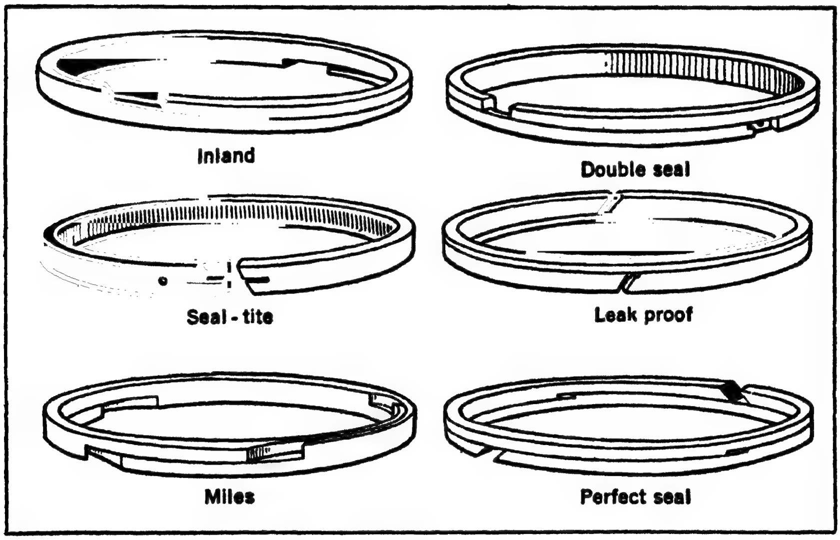


Fig. 328.—Diagrams Detailing Unconventional Piston Rings Suitable for Wide Grooves Only.

Light Test for Piston Rings.—A test which has been much used in the past in the piston ring industry is the light test. The ring is inserted into a ring gauge of a bore equal to that of the cylinder for which it is intended, and an electric lamp is then placed behind the gauge. If no light shines through between the ring and the gauge at any point of its circumference, the ring is considered to be perfect. This test has some merit, and a ring which passes it is at least a better ring than one which does not. However, it is not conclusive, for a ring may make contact with the gauge at a particular point and yet not exert any pressure on it. Thus a ring which passes this test and which exerts an average pressure against the cylinder-wall of one pound per inch in length of the ring, may exert a pressure of one-half pound over a certain one-quarter-inch section of length and no pressure at all over another similar section. According to Mr. Heldt the

light test serves to separate very faulty rings from those which are tolerably good, but it is not sufficiently sensitive to distinguish the moderately good from very good rings.

Piston Ring Joints.—A practical arrangement of the gap, which prevents injury to the ring from butting, has been adopted by several manufacturers and is known as the diagonal cut. This consists in cutting a section out of the ring whose faces make an angle of, say, 45 degrees with a radial line. Then, if the ring expands so much in service that the ends begin to butt, one end is forced under the other down into the groove. The length of gap is usually made either 0.0025 or 0.003 inch per inch of bore. There seems to be an impression among engineers that the lap joint is slightly superior to the diagonal joint, but inasmuch as the joint is practically closed up when the engine is working at normal temperature, it is hard to see why there should be any material difference between the two. With rings which are turned to the diameter of the bore and then put under tension by peening or some similar method, it is practically necessary to use the diagonal joint, as the lap joint requires the removal of a considerable section of the ring, which is not permissible if the rings are turned to the diameter of the bore in the first place. When a lap joint is used, the ring is rough-turned enough larger than the cylinder bore to allow for the metal removed in making the lap. The rings are then clamped in a fixture with the gap at the point where it will be when the ring is heated and then it is finished by grinding to make a ring that will bear all around the cylinder. Various fancy joints have been made, such as shown at Fig. 324 but these are practical only in wide rings now seldom used in aviation engines.

Oil Rings.—In many engines the supply of lubricant to the cylinders is excessive and a great deal of oil gets by the pistons and is burned in the combustion-chamber, causing a smoky exhaust and carbon deposits. For the proper distribution of oil over large bearing surfaces oil grooves have always been considered of great importance. Most of the oil is thrown onto the cylinder-wall when the piston is near the top end of the stroke, and if the bottom ring is provided with a sharp edge at the bottom and a groove is cut directly below the bottom ring groove, the ring will scrape the oil off the cylinder bore into this groove, from which it can generally return to the crankcase through small holes drilled through the wall of the piston at the oil groove.

Instead of having an oil groove in the piston, it can be cut in the ring itself. Some cut a substantially square slot in the ring at its lower outer edge and in some cases radial slots are provided on the lower side of the ring through which the oil thus scraped off may pass to the bottom of the ring groove and hence back to the crankcase. Others cut the oil groove in the middle portion of the ring, away from the edge, and in some cases the top edge of the ring is beveled off so it will not tend to scrape the oil off the cylinder bore during the upward stroke of the piston. Various methods of oil grooving are shown at Fig. 325 B. As a rule, manufacturers of oil wiper rings recommend the use of only one such ring per piston, in the lowest groove above the piston pin. A very good form of oil control ring is shown at Fig. 325 A. In this construction, only possible with rings at least 3/16 inch wide the oil return passages are machined from the outer

to the inner faces of the ring and provide a ready passage for the oil to flow back into the piston interior through the holes in the piston wall at the ring groove provided for the purpose.

Quick Seating Rings.—In spite of the most accurate machine work on both the cylinder bore and the rings, the latter never fit the bore perfectly when first installed, and for a certain period after a new engine is put into service, or after new rings have been placed on the pistons, the power and efficiency of the engine will increase. It is, of course, desirable to reduce this "running-in" period to a minimum; that is, to so build the engine that the maximum output and the maximum efficiency are attained with the least delay.

Formerly, piston rings were generally ground on the outside surface, the surface which bears against the ground cylinder-wall. It has been found, however, that if the outside surface of the ring is turned in the lathe as shown at Fig. 326 A, it seats itself to the cylinder-wall very much quicker, and most manufacturers of piston rings now turn their rings, some even taking a comparatively coarse cut. The turning operation, instead of giving the smooth finish obtained by grinding, provides the surface with what is practically a very fine screw thread. By using a tool of the proper shape this thread can be made of very small width as compared with the spaces between threads, and will then wear off rapidly. Mr. Heldt states that some manufacturers get the same effect of rapid seating by undercutting one-half or two-thirds of the surface of the ring to a depth of about 0.002 inch as shown at Fig. 326 C. The total pressure of the ring being then concentrated on the reduced contact surface, this surface wears down faster than it otherwise would, or at least that is the theory. One manufacturer even provides his rings with three concentric surfaces on the outside, each about 0.0015 inch below the one of next larger radius, and as the higher surfaces wear down the others come into effect successively. This ring is shown at Fig. 326 B.

Those who still grind their rings on the outside raise the objection to the quick-seating principle, that as the diameter is reduced by the wear of the rough exterior surface, the gap is increased in length at a rate 3.1416 times faster. With any of the ordinary forms of gap, there is, of course, a chance for leakage, but the leakage area at any particular gap in the worst case is only the product of the length of the gap into the clearance between the piston and the cylinder. For instance, if the piston clearance at the ring belt is 0.012 inch and the gap is increased in length by 0.006 inch by the wear of the turned face, then the leakage area increases by 0.000072 square inch and not very much gas will leak through this infinitesimal area during the small fraction of a second occupied by the compression or explosion of the charge.

Machining Ring Grooves.—In order that rings may perform their function properly they must be accurately fitted into the ring grooves. The grooves should be machined perfectly smooth and true with the axis of the piston. Some makers roll the grooves to size. Generally a side clearance of 0.001 inch in the groove is allowed, but some makers go beyond this and hold to clearance limits of 0.00025 and 0.00075, by selective assembling. In inspecting rings the following items are generally checked: Width of

ring, on which the tolerances are usually 0.0005 inch; out-of-roundness when compressed by a light wire or cord, maximum tolerance 0.010 inch; tangential pressure required to close ring ¾ to one pound per inch diameter for a 3/16-inch ring and in proportion for rings of different width.

Leak-Proof Piston Rings.—In order to reduce the compression loss and leakage of gas by the ordinary simple form of diagonal or lap joint one-piece piston ring a number of compound rings have been devised and are offered by their makers to use in making replacements. The leading forms are shown at Fig. 327. That shown at A is known as the "Statite" and consists of three rings, one carried inside while the other two are carried on the outside. The ring shown at B is a double ring and is known as the McCadden. This is composed of two thin concentric lap joint rings so disposed relative to each other that the opening in the inner ring comes opposite to the opening in the outer ring. The form shown at C is known as the "Leektite," and is a single ring provided with a peculiar form of lap and dove tail joint. The ring shown at D is known as the "Dunham" and is of the double concentric type being composed of two rings with lap joints which are welded together at a point opposite the joint so that there is no passage by which the gas can escape. The Burd high compression ring is shown at E. The joints of these rings are sealed by means of an H-shaped coupler of bronze which closes the opening. The ring ends are made with tongues which interlock with the coupling. The ring shown at F is called the "Evertite" and is a three-piece ring composed of three members as shown in the sectional view below the ring. The main part or inner ring has a circumferential channel in which the two outer rings lock, the resulting cross-section being rectangular just the same as that of a regular pattern ring. All three rings are diagonally split and the joints are spaced equally and the distances maintained by small pins. This results in each joint being sealed by the solid portion of the other rings.

The use of a number of light steel rings instead of one wider ring in the groove is found on a number of automobile powerplants, but as far as known, this construction is not used in airplane powerplants. It is contended that where a number of light rings is employed a more flexible packing means is obtained and the possibility of leakage is reduced. Rings of this design are made of square section steel wire and are given a spring temper. Owing to the limited width the diagonal cut joint is generally employed instead of the lap joint which is so popular on wider rings.

Compound and Unusual Piston Rings.—In automobiles and similar automotive vehicles, where rings of 3/16 inch and ¼ inch width have been used, various forms of compound and two- and three-piece rings such as illustrated at Fig. 328 have been used. Compound rings are not well adapted to engines used for aircraft because they are not as reliable as the simpler rings and besides, it is very difficult to work out a successful design to fit narrow grooves. Mr. P. M. Heldt describes one of these compound rings in which there are two cast-iron rings at opposite sides of the groove, with a flat steel spring with radial corrugations in it between them. In order to get the compound ring into the groove, the three parts have to be pressed together endwise, and once in the groove the steel ring forces those of cast iron against the sides of the groove. In most rings of this type the

contact surface between two members is inclined to the axis of the ring, and the radial pressure of one member creates both a lateral and a radial pressure in the other. One member is usually depended upon to effect the seal on the circumference and at one side of the groove, the other at the remaining side of the groove.

Reports on the efficiency of such rings are at variance with each other. It is generally conceded that they work well while new, though it is claimed by some that even then the friction of the ring against both sides of the groove exerts a certain damping action which makes it difficult for the ring to follow any inaccuracies of the bore, such as a slight tapering from end to end. Opponents of this construction say that rings of this type have no "life." It is, of course, quite conceivable that when rings of this class, comprising two or more pieces, are covered with carbon and gummed up, they are not as efficient as when new and clean, but this applies to all rings but to a lesser degree in one-piece rings. Perhaps the best proof of the efficiency of such rings lies in the rapid increase in the number of different designs on the market and their use in automobiles. The fact that laterally expanding rings are not as free in the groove as single-piece rings is admitted by the manufacturers of such rings, but they consider this to be an advantage rather than a fault, claiming that it tends to prevent piston slap. Some of the rings, such as the Inland, are a one-piece ring with a greatly exaggerated diagonal expansion slot. Others, such as the Double Seal and Leak Proof, are assembled of independently machined members. Narrow ring grooves, as used in aircraft engine pistons are not favorable to the use of compound rings.

Keeping Oil Out of Combustion-Chamber.—An examination of the engine design that is economical in oil consumption discloses the use of tight piston rings, large centrifugal oil throw rings on the crankshaft where it passes through the case, ample cooling fins in the pistons, vents between the crankcase chamber and the valve enclosures, etc. Briefly put, cooling of the oil in this engine has been properly cared for and leakage reduced to a minimum. To be specific regarding details of design: Oil surplus can be kept out of the explosion chambers by leaving the lower edge of the piston skirt sharp and by the use of a shallow groove (C), Fig. 329, just below the lower piston ring. Small holes are bored through the piston walls at the base of this groove and communicate with the crankcase. The similarity of the sharp edges of piston skirt (D) and piston ring to a carpenter's plane bit, makes their operation plain.

The cooling of oil in the sump (A) can be accomplished most effectively by radiating fins on its outer surface, though this calls for the use of an aluminum casting instead of a pressed steel oil pan, now used on most automotive engines produced in quantities. The lower crankcase should be fully exposed to the outer air. A settling basin for sediment (B) should be provided having a cubic content not less than one-tenth of the total oil capacity as outlined at Fig. 329. The depth of this basin should be at least 2½ inches, and its walls vertical, as shown, to reduce the mixing of sediment with the oil in circulation. The inlet opening to the oil pump should be near the top of the sediment basin in order to prevent the entrance into the pump with the oil of any solid matter or water condensed from the

products of combustion. This sediment basin should be drained after every five to seven hours air service of an airplane engine. Concerning filtering screens there is little to be said, save that their areas should be ample and the mesh coarse enough (one-sixteenth of an inch) to offer no serious resistance to the free flow of cold or heavy oil through them; otherwise the oil in the crankcase may build up above them to an undesirable level. The necessary frequency of draining and flushing out the oil sump differs greatly with the age (condition) of the engine and the suitability of the oil used. In broad terms, the oil sump of a new engine should be thoroughly drained and flushed with kerosene at the end of the first 200 miles, next at the end

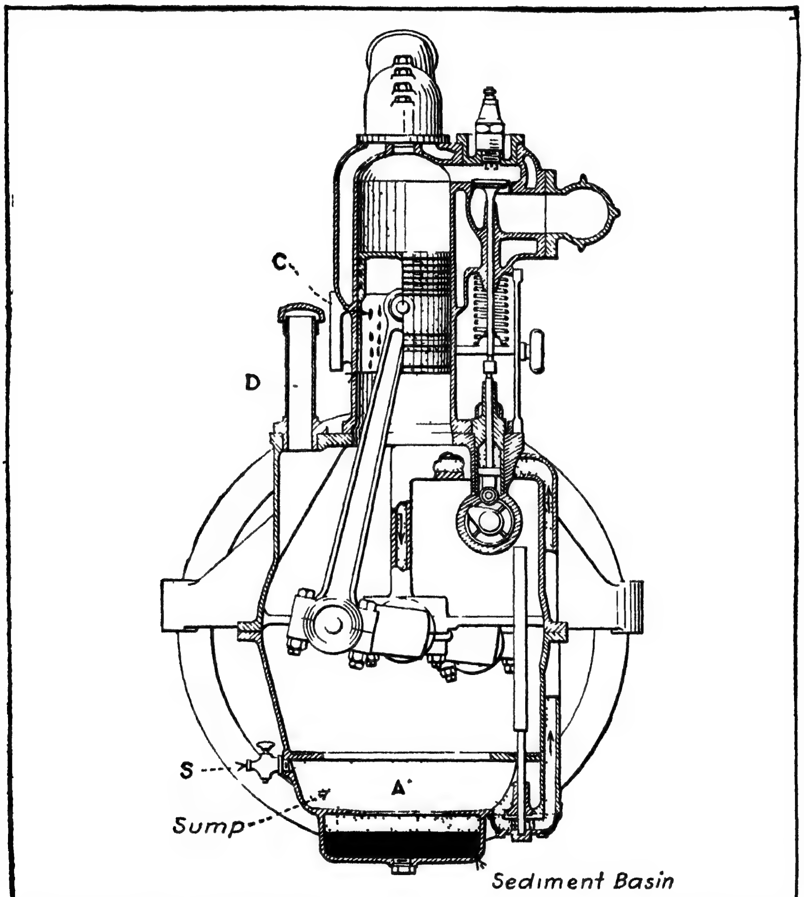


Fig. 329.—Sectional View of Automobile Engine Showing Method of Preventing Oil Leakage by Piston Rings.

of 500 miles and thereafter every 1,000 miles. While these instructions apply specifically to automobile motors, it is very good practice to change the oil in airplane engines frequently. In many cases, the best results have been secured when the oil supply is completely replenished every five hours that the engine is in operation.

Connecting Rod Forms.—The connecting rod is the simple member that joins the piston to the crankshaft and which transmits the power imparted to the piston by the explosion so that it may be usefully applied. It transforms the reciprocating movement of the piston to a rotary motion at the crankshaft. A typical connecting rod group for a Vee-engine and wristpins are shown at Fig. 330 A. It will be seen that it has two bearings, one at either end. The small end is bored out to receive the wristpin which joins it to the piston, while the large end has a hole of sufficient size to go on the crankpin. The airplane and automobile engine connecting rods are

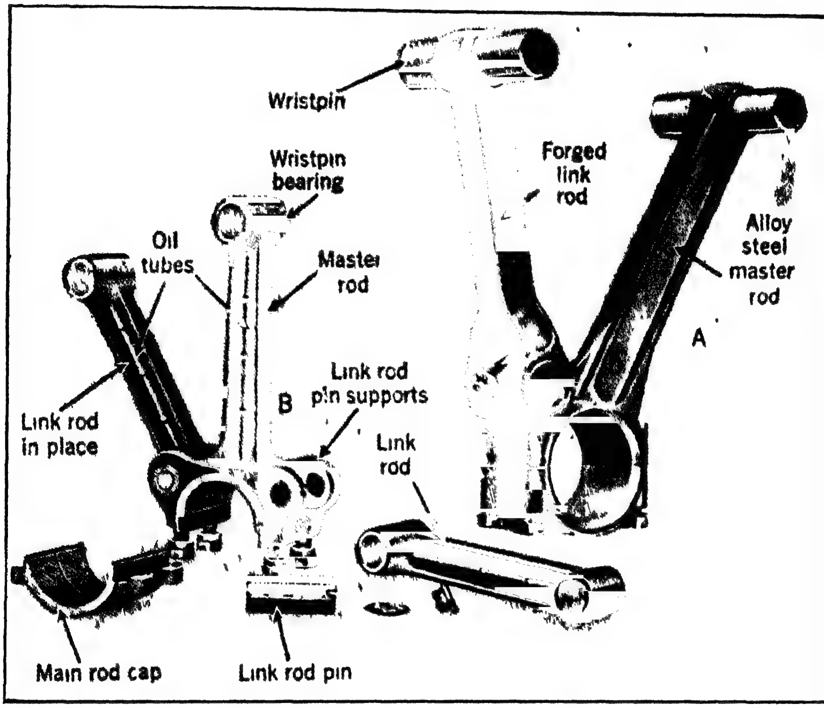


Fig. 330.—Illustration Showing Master and Link Rod Assembly of Good Modern Design. A—Connecting Rod Assembly Used on Fiat A20 "Vee" Engine. B—Master Connecting Rod and Two Link Rods Employed on Napier-Lion W Type Engine.

nearly always steel forgings though duralumin forgings have also been used successfully as well as built up tubular rods. In marine engines it is sometimes made a steel or high tensile strength bronze casting. In all cases it is desirable to have softer metals than the crankshaft and wristpin at the bearing point, and for this reason the connecting rod is usually provided with bushings of anti-friction or white metal at the lower end, and bronze at the upper. The upper end of the connecting rod may be one piece, because the wristpin can be introduced after it is in place between the bosses of the piston. The lower bearing must be made in two parts in all cases where one-piece crankshafts are used because the crankshaft cannot be passed through the bearing owing to its irregular form. The rods of the Gnome rotary engine were all one-piece types, as shown at Fig. 127, owing

to the construction of the "mother" rod which receives the crankpins. The complete connecting rod assembly is shown in Fig. 311, also at Fig. 331 A. The "mother" rod, with one of the other rods in place and one about to be inserted, is shown at Fig. 331 B. The built-up crankshaft which makes this construction feasible is shown at Fig. 331 C. The "mother" or "master" rod assembly with its link rods is a common type for radial engines because any number of rods, up to nine, can be conveniently grouped around a central crankpin. The grouping of the connecting rods around a master rod in the Wright J5 Whirlwind motor is clearly shown at Fig. 332, which outlines a cutaway section of the engine.

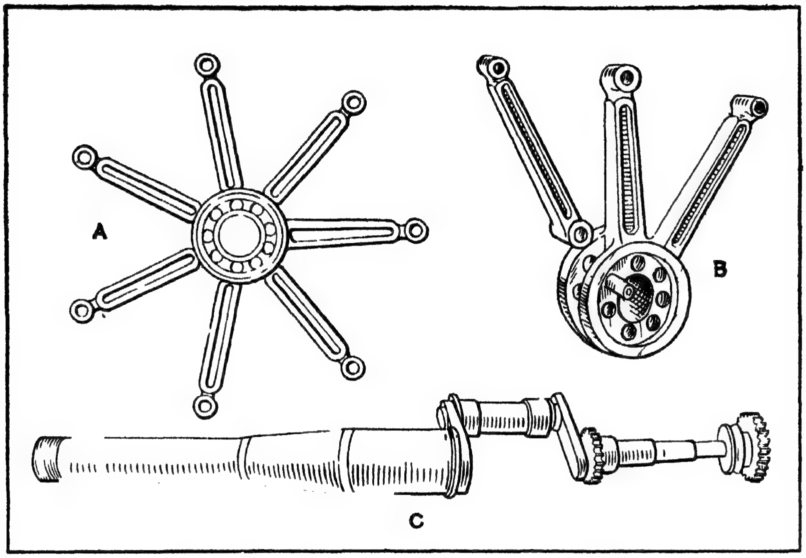


Fig. 331.—Connecting Rod and Crankshaft Construction of Early Gnome "Monosoupape" Engine.

Some of the various designs of connecting rods that have been used in motors of various types are shown at Fig. 333. That at A is a simple form often employed in single-cylinder motorcycle motors, having built-up crankshafts. Both ends of the connecting rod are bushed with a one-piece bearing, as it can be assembled in place before the crankshaft assembly is built up. In modified form this is the type of rod used as a link rod in radial engines, but the upper and lower ends are usually the same diameter or bore. The pattern shown at B is one that has been used to some extent on heavy work, and is known as the "marine type." It is made in three pieces, the main portion being a steel forging having a flanged lower end to which the bronze boxes are secured by bolts. The construction would be much too heavy for aviation engines. The modified marine type depicted at C is the form that has received the widest application in automobile and aviation engine construction where four or six cylinders are in line. It consists of two pieces, the main member being an alloy steel or dural forging having the wristpin bearing and the upper crankpin bear-

ing formed integral, while the lower crankpin bearing member is a separate forging secured to the connecting rod by bolts. In this construction bushings of anti-friction metal are used at the lower end, and a bronze bushing is forced into the upper or wristpin end. The rod shown at D has been widely used on stationary engines and some early automobile engines. It is similar in construction to the form shown at C, except that the upper end is split in order to permit of a degree of adjustment of the wristpin bushing, and the lower bearing cap is a hinged member which is retained by one bolt instead of two. When it is desired to assemble it on the crankshaft the lower cap is swung to one side and brought back into place when

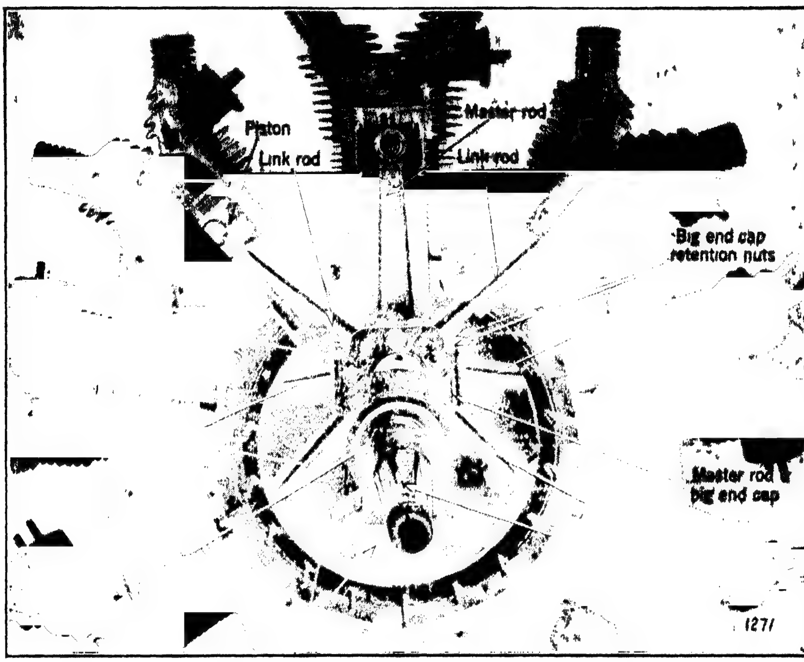


Fig. 332.—Part Sectional View of Wright "Whirlwind" Motor, with Crankcase Cover Removed, Showing Construction of the Master Connecting Rod and Link Rod Assembly, Also Method of Installation on One Piece Crankshaft Made Possible by Split Big End Design. Note Safety Wiring of Connecting Rod Cap Clamp Bolts.

the connecting rod has been properly located. Sometimes the lower bearing member is split diagonally instead of horizontally, such a construction being outlined at E, but this is seldom found in modern engines of any type.

In a number of instances, instead of plain bushed bearings anti-friction forms using ball or rollers have been used at the lower end. A ball-bearing connecting rod is shown at F. The big end may be made in one piece, because if it is possible to get the ball-bearing on the crankpins it will be easy to put the connecting rod in place. Ball-bearings are not used very often on connecting rod big ends because of difficulty of installation and the severe loading, though when applied properly as main crankshaft bearings they give satisfactory service and reduce friction to a minimum.

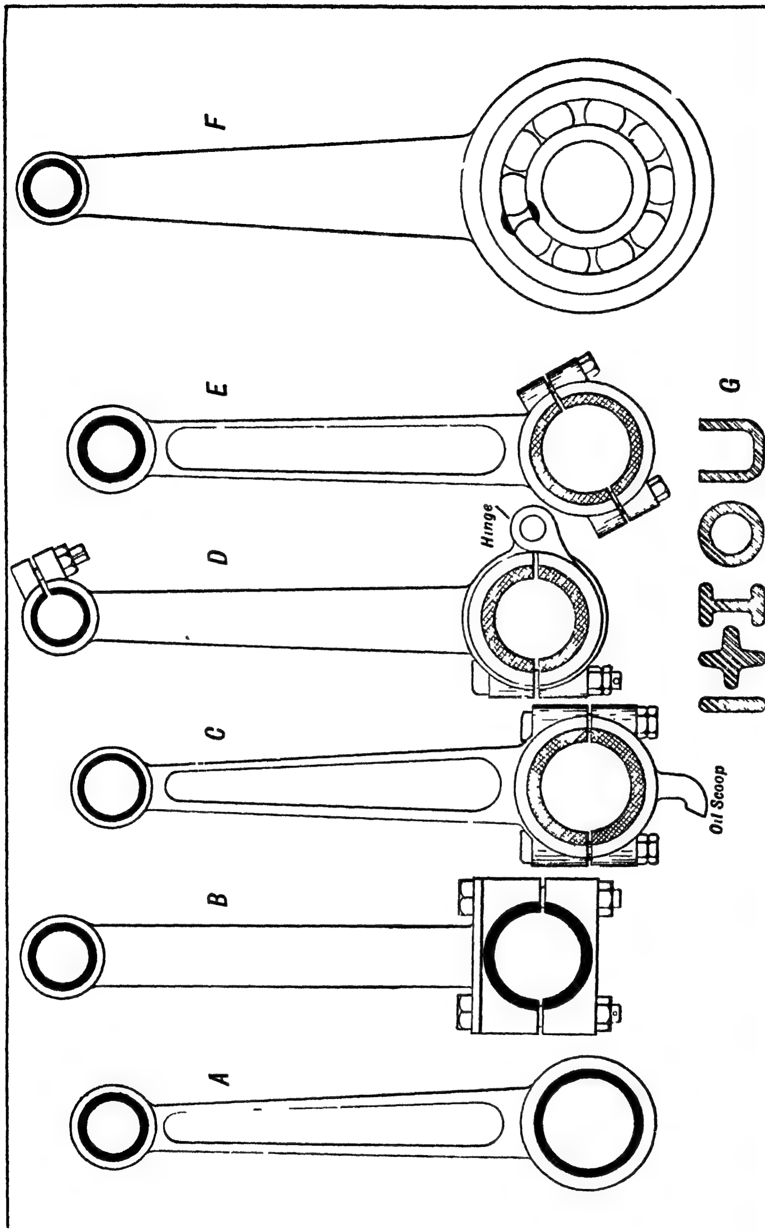


Fig. 333.—Connecting Rod Types of Automotive Engines Summarized. A—Connecting Rod Forged in One Piece, Used in Small Engines Having Built-Up Crankshafts. B—The Marine Type is a Popular Form on Heavy Engines. C—Conventional Automobile Type is a Modified Marine Form. D—Rod Having Hinged Lower Cap and Split Wristpin Bushing, Designed to Permit Easy Adjustment, is Suited for Slow-Speed Stationary Engines. E—Connecting Rod Having Diagonally Divided Big End. F—Connecting Rod with Anti-Friction Bearing in the Big End. G—Sections of Connecting Rods Showing Structural Shapes Commonly Employed.

One of the advantages of the ball- or roller-bearing is that it requires no adjustment, whereas the plain bushings depicted in the other connecting rods must be taken up from time to time to compensate for wear.

This can be done in forms shown at B, C, D, and E by bringing the lower bearing caps closer to the upper one and scraping out the babbitt metal lining to fit the shaft. A number of liners or shims of thin brass or

copper stock, varying from .002 inch to .005 inch, are sometimes interposed between the halves of the bearings of automobile connecting rods when first fitted to the crankpin. As the brasses wear the shims may be removed and the portions of the bearings brought close enough together to take up any lost motion that may exist, though in most aviation engines no shims are provided and depreciation can be remedied only by installing new linings and scraping to fit.

Connecting Rod Sections.—The various structural shapes in which connecting rods are formed are shown in section at G. Of these the I section is most widely used in airplane engines, because it is strong and a very easy shape to form by the drop-forging process or to machine out of the solid bar when extra good steel is used. Where extreme lightness is desired, as in small high-speed motors used for cycle propulsion, the section shown at

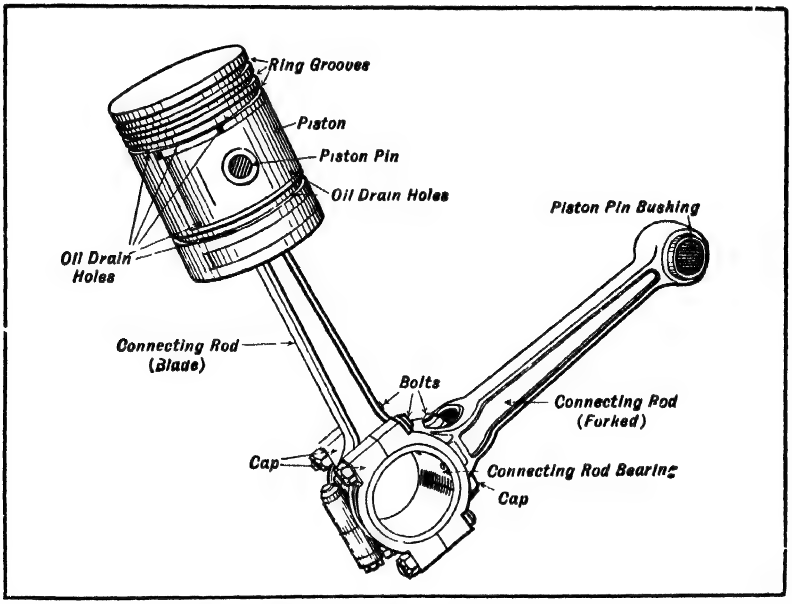


Fig. 334.—Double Connecting Rod Assembly Using Blade and Fork Rods for Installation on Single Crankpin, When Opposite Cylinders Have the Same Center Lines.

the extreme left is often used. If the rod is a cast member as in some marine engines, the cross, hollow cylinder, or U sections are sometimes used. If the sections shown at the right are employed, advantage is often taken of the opportunity for passing lubricant through the center of the hollow round section on vertical motors or at the bottom of the U section, which would be used on a horizontal cylinder powerplant. It will be apparent that any hollow section structural shape could be employed in built-up rods for aviation engines and it often is. Forged steel end pieces are electrically butt welded to an alloy steel connecting tube and a very light and strong rod is obtained that gives the added advantage of providing an oil passage through its center.

Connecting Rods for Vee Engines.—Connecting rods of Vee engines have been made in two distinct styles. The forked or "scissors" joint rod assembly is sometimes employed when the cylinders are placed directly opposite each other. The "blade" rod, as shown at Fig. 334, fits between the lower ends of the forked rod, which oscillate on the bearing which encircles the crankpin. The lower end of the "blade" rod is usually attached

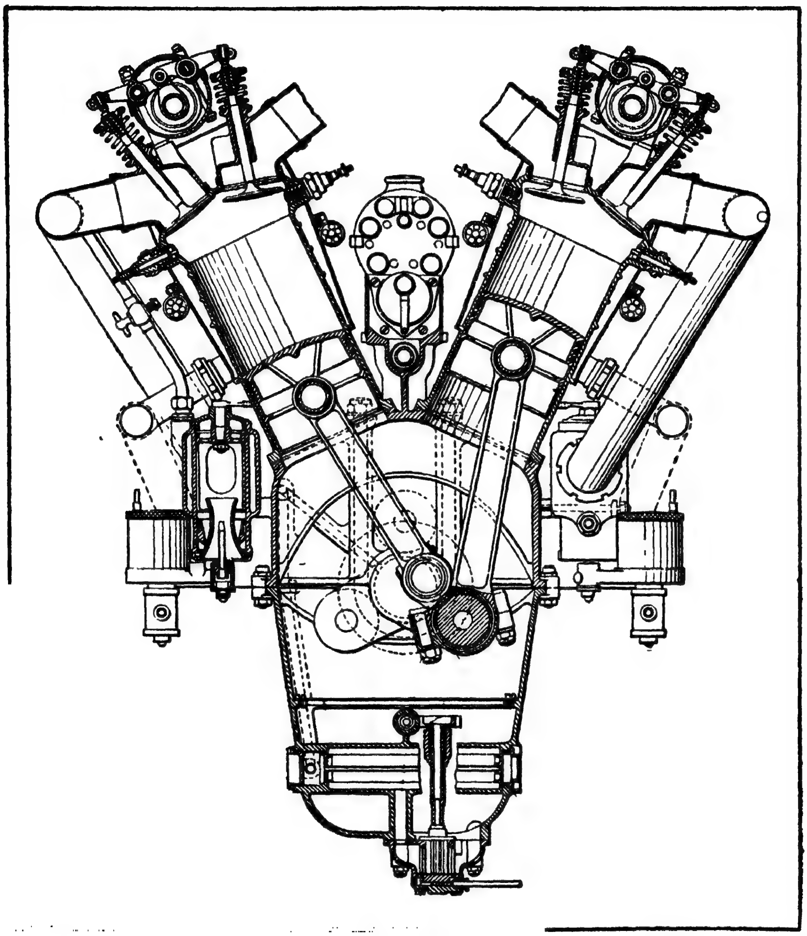


Fig. 335.—Part Sectional View of Early Renault Twelve-Cylinder Water-Cooled "Vee" Engine Showing Connecting Rod Construction and Other Important Internal Parts.

to the bearing brasses, the ends of the "forked" rod move on the outer surfaces of the brasses. Another form of rod devised for use under these conditions is shown at Fig. 330 and installed in an aviation engine at Fig. 335. In this construction the shorter rod is attached to a boss on the master rod by a short pin to form a hinge and to permit the short rod to oscillate as the conditions dictate. This form of rod can be easily adjusted when the bearing depreciates, a procedure that is difficult with the forked type rod. Another practice is to stagger the cylinders and use side-by-side rods as is

done in the Curtiss OX5 engine. Each rod may be fitted independently of the other and perfect compensation for wear of the big ends is possible.

Rods for W and X Engines.—In engines where three banks of cylinders are used, as in W engines or where four banks are utilized, as in the Packard X engine, it will be apparent that any staggering of cylinders to place the rods side by side would be very difficult and would call for crank-pins of considerable length and make an unwieldy and long engine. For

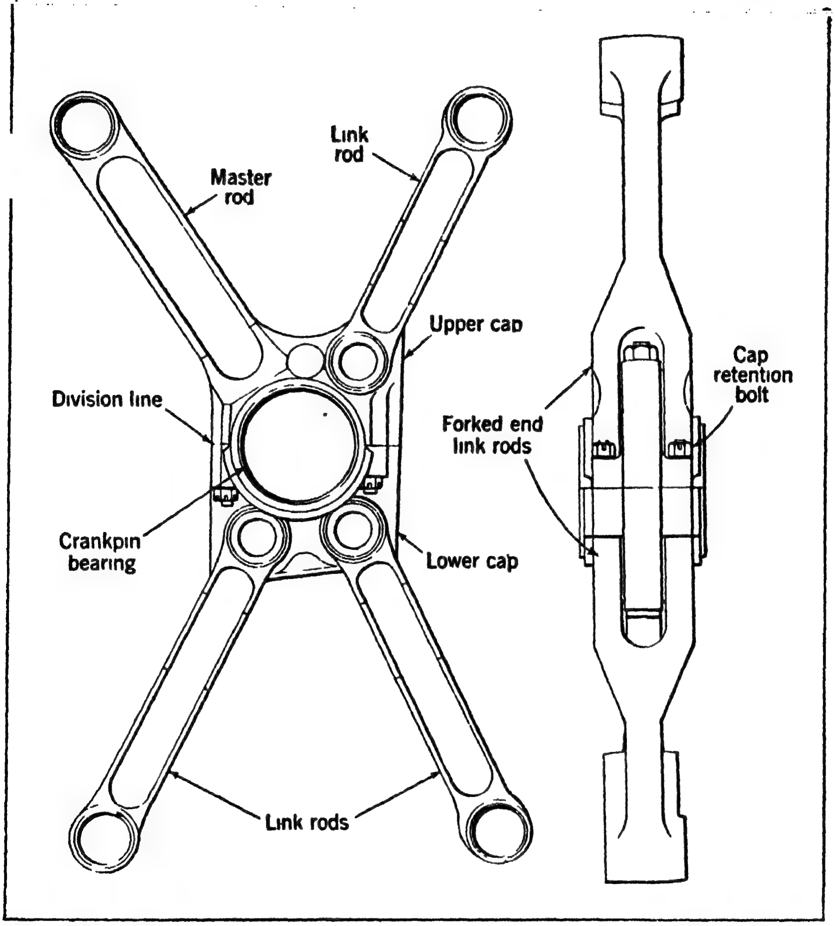


Fig. 336.—The Unconventional but Substantial and Practical Connecting Rod Assembly Designed for the Packard Twenty-Four-Cylinder X Engine. Note that the Pressure of Four Pistons is Received by One Crankpin Bearing.

this reason, the connecting rod arrangement shown at Fig. 230 B is used on Napier Lion engines. The main rod has two sets of ears to support the lower end of the link rods. The unusual design of the Packard connecting rod assembly shown at Fig. 336 is necessary because four banks of cylinders in X arrangement are used. This comprises a master rod forging with the upper bearing cap forged integral, the lower bearing cap being bolted to it. The link rods are yoked and fit lugs extending from the bearing caps.

Rods for Radial Cylinder Engines.—When radial cylinder engines are used, the connecting rod arrangement is very simple and in any single row engine or single crankpin design, all center lines may coincide, i.e., all cylinders may be on one plane, no offsetting from front to rear being neces-

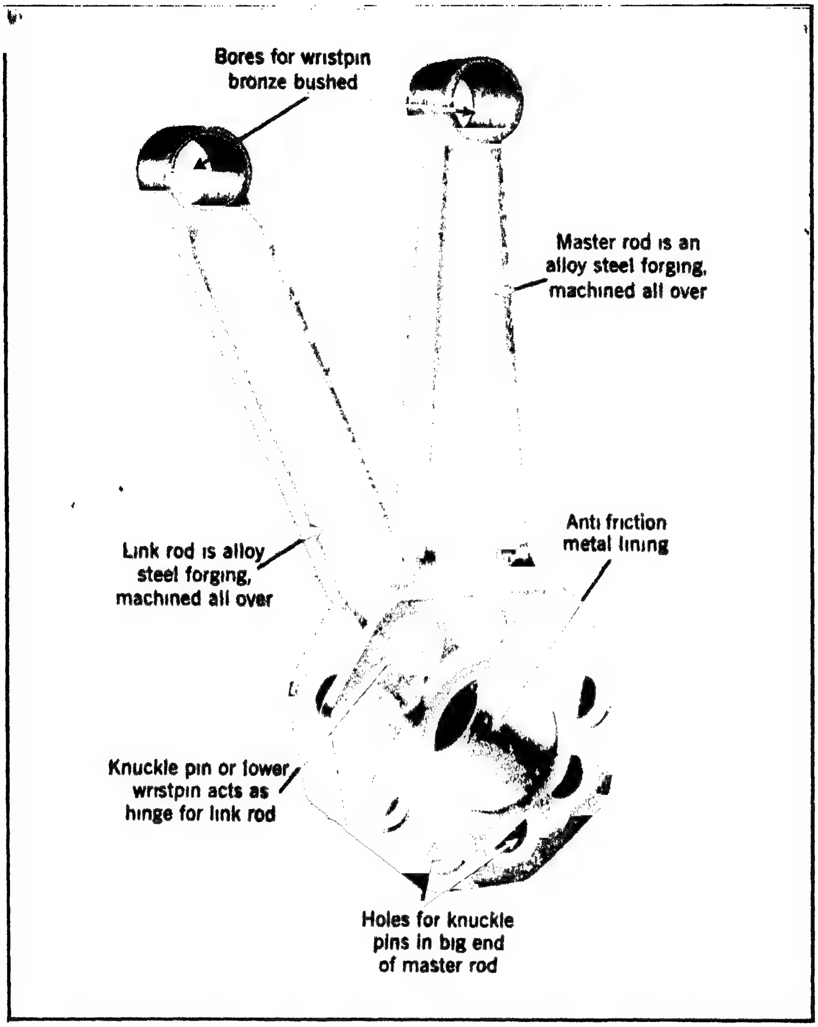


Fig. 337.—The Master Connecting Rod Used on the Pratt & Whitney "Wasp" Engine Has a Solid Big End Bearing. One of the Link Rods is Shown Installed to Show Relation in the Assembly. Seven Other Link Rods Radiate from the Big End, but These are Not Shown in this Illustration.

sary. Two methods of rod construction are followed, these relating to the master rod. The big end may be a split bearing type as in the Wright Whirlwind illustrated at Fig. 332 in which a one-piece crankshaft is used or the connecting rod may be a one-piece construction and the crankshaft made in two pieces to permit assembling the big end on the crankpin. The

Pratt & Whitney Wasp engine uses a built-up shaft and a one-piece big end, which is shown clearly at Fig. 337. The use of a single-piece rod permits of higher engine speeds than are possible with the divided big end construction and makes possible the use of geared engines as in the Bristol Jupiter, which is a large radial air-cooled engine running at high speed.

The limitations imposed on rotative speed by split rod ends are recognized by radial engine designers but in direct drive engines, the propeller efficiency is really the limiting factor and split big ends on master rods, as in the Whirlwind engines do not limit the speed which is as high as efficient

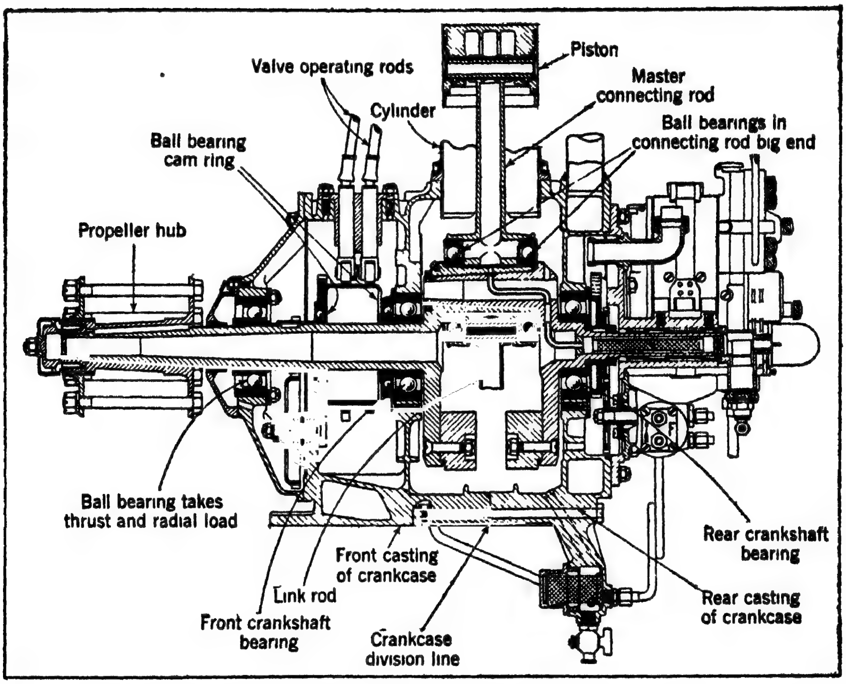


Fig. 338.—Sectional View of the Crankcase of the Siemens Engine, Showing Extensive Use of Anti-Friction Bearings for Carrying Crankshaft, Cam Ring, and Connecting Rod Big End Bearing Loads.

propeller designs can utilize when mounted directly on the crankshaft. In a discussion on the subject before the S. A. E., Mr. C. L. Lawrance, internationally famous as the designer of the Whirlwind engines stated he believed that engineers have only just begun developing speed possibilities of radial cylinder air-cooled engines. He admitted that with the type of connecting rod held together by four bolts many difficulties undoubtedly would develop at unduly high engine-speed, such as scuffing of the parts and breakage of the bolts, but with a one-piece connecting rod, in which no localization of stresses occurs due to the forces having to pass through the bolts, the construction is much stronger than the standard connecting rod of a water-cooled engine. Mr. Lawrance doubted very much whether any more trouble due to connecting-rod failure will occur in running a radial engine up to high speeds than occurs with water-cooled engines. Experi-

ence has shown that we can run them up to somewhat high speeds or 2,200 r.p.m. and he thinks we have only begun to develop the possibilities. The loading on the bearing of a radial engine crankpin does not remain in the same place but travels around the circumference of the bearing, so that during the nonexplosion strokes and two revolutions of the crankshaft the load has traveled twice around the bearing. If proper cooling is furnished for that bearing, no reason exists for its failure, whereas in a Vee-type engine the load is sustained in much the same place all the time.

The design of connecting rod shown at Fig. 337 offers several advantages that make it well suited to high-speed engines. In the first place, by pouring the babbitt or alloy bearing metal directly into the solid big end, excellent heat conductivity is obtained. The holes for the lower bearing pins of the radial link rods can be brought closer to the bearing than in a bolted big end construction and the weight of the big end assembly correspondingly reduced.

Ball and Roller Bearing Connecting Rods.—Anti-friction bearings have long been used to a limited extent in connecting rod heads—in recent years particularly in engines of racing cars. The majority of motorcycle engines built in this country have such bearings, and a touch of timeliness is lent the subject by the announcement that the Maybach engines of the ZR3 have only roller-bearings on the crankshaft. The radial cylinder air-cooled engines also use ball-bearings for crankshaft support and in some cases, as exemplified by the Siemens engine, ball-bearings are also used in the big end bearing of the master connecting rod as shown at Fig. 338.

An article embodying an analysis of the problem referred to and a review of the proposed and applied solutions was recently published by J. Dauben in *Der Motorwagen,* and the following information on the subject is abstracted from that article. He first refers to the rather unfavorable conditions under which the roller-bearing has to work in the connecting rod head. The rollers in such a bearing have a dual motion, including a rotary motion around the axis of the crankpin and a planetary motion around the axis of the crankshaft. They are subjected to centrifugal forces due to both of these motions, and the resultants of these centrifugal forces on the rollers in the different positions are indicated in Fig. 339 A. These forces are of a magnitude which is far from being negligible, and in this connection it may be mentioned that at 3,000 r.p.m. and a stroke of four inches each roller is subjected to a force practically 500 times its own weight, pressing it outward. In addition, the cage is subjected to a similar force which is transmitted to the rollers and must add materially to the friction. Another factor which has an unfavorable influence on the operation of a roller-bearing in the connecting rod head is that the rolling motion of the roller is nonuniform, due to the oscillating motion of the rod itself. This involves accelerations and decelerations of both the rollers and the cage, and causes considerable friction between roller and cage.

Owing to the fact that the forces which have such an unfavorable influence on the rollers and cages are directly proportional to the weights of the parts, these parts must be held as small as possible. This also is the reason why ball-bearings are out of the question for this purpose in very high-speed engines as for equal carrying capacity balls are materially

heavier than rollers. It is advisable to keep the rollers down in diameter ($\frac{1}{4}$ to $\frac{5}{16}$ inch), without, however, overtaxing their carrying capacity. It is also good practice, especially in view of the nonuniform motion of the connecting rod head bearing, to keep the diameters of the races as small as possible and for this reason it is usually best to do without special inner races and let the rollers run directly on the hardened crankpin. Ball-bearing manufacturers usually advise against this practice, because they prefer to see standard bearings used, but experience has shown that where

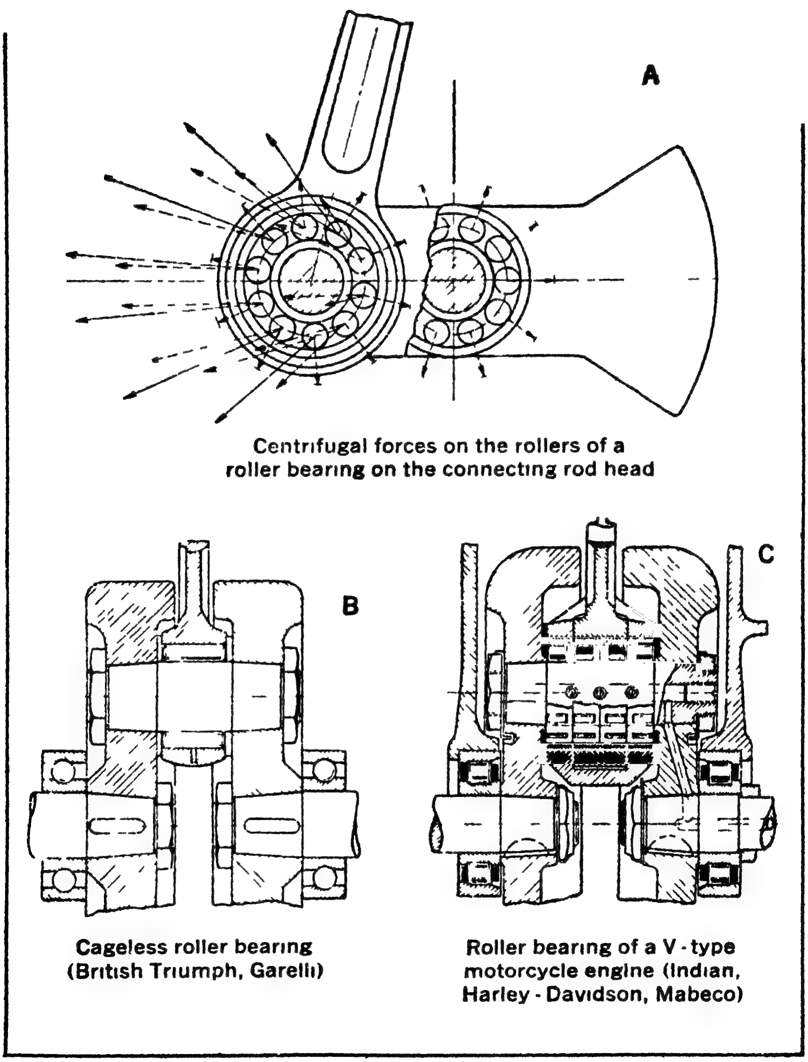


Fig. 339.—View at A Shows the Centrifugal Forces on the Roller Bearing of a Connecting Rod Head. B—Cageless Roller Bearing Used on Triumph (British) Motorcycle Engine. C—Roller Bearing Design on "Vee" Type American Motorcycle Engines.

the proper grade of steel is selected and it is suitably hardened the crankpin can be used for a roller race without hesitation.

Great importance attaches to the use of rigid and accurate cages. So far massive cages of a tough bronze have given the best service. However, since reduction of weight is essential, for the reasons enlarged upon in the foregoing, the use of light metal, and especially of magnesium alloy, is promising, even though the results obtained with it have not always been satisfactory. With one-piece crankshafts it is necessary to make the cage in four parts, which must be so designed as to overlap one another and must be bolted or riveted together. The cage must be assembled directly on the crankshaft, which is a great hindrance in quantity production. An all around satisfactory design for a roller-bearing connecting rod head for automotive engines has not been developed as yet, but some progress in this direction is noticeable, especially in the field of small engines.

In order that roller-bearing connecting rod heads may give satisfactory service it is essential that a reliable oil feed be provided for them. In so-called moderate duty engines, such as those of stock automobiles, splash lubrication may suffice, but it certainly does not in racing engines, which often must run at high speed under full throttle for hours at a time. The only reliable method of feeding oil to the connecting rod bearings of aviation engines is by pressure through a drilled crankshaft. This system has proved the most dependable in connection with smooth bearings, and as there is considerable sliding friction also in connecting rod roller bearings these bearings should be as thoroughly lubricated as smooth bearings.

Roller Separators Important.—The simplest form of connecting rod head roller bearing is that without cage, as represented in Fig. 339 B. The rollers are of small diameter and comparatively long, and they run directly on the hardened crankpin and in the connecting rod head. It is important that they fill up the space with only very little clearance so that they will not hammer against each other with great force and cannot "cock." The only disadvantage of this type of bearing is that the rollers have a tendency to "cock" and break and then generally destroy the entire bearings. This cageless connecting rod bearing has given excellent service in a number of British and American cycle engines. An improvement upon the bearing just described is represented by that shown in Fig. 339 C which also is used in cycle engines (of the Vee type). Here also the rollers run directly on the hardened crankpin, while the outer races are pressed into the connecting rod head. The rollers, which are ¼ inch in diameter and $\frac{5}{16}$ inch long, are inserted into solid bronze cages, of which there are two for each rod. One side of the cage is open and the different cages are so arranged that an open and a closed side are always together. At the ends there are hardened steel discs, the same as in the design shown at A. Lubrication through the pin is provided for, but the radial holes are not located in the path of the rollers but between them, as that would result in the rollers wearing depressions in the bearing surface owing to the reduced bearing surface.

Method of Using Standard Bearings.—Another example from motorcycle practice is shown in Fig. 340 A. Standard roller bearings are used, the outer races being a free fit in the unhardened rod, so they will slowly

travel around and distribute any wear over the whole surface. In order to hold the connecting rod against lateral forces, a wire ring is let into a groove turned into the two outer races. An interesting feature consists of the spring steel conical washers which hold the inner races in an axial

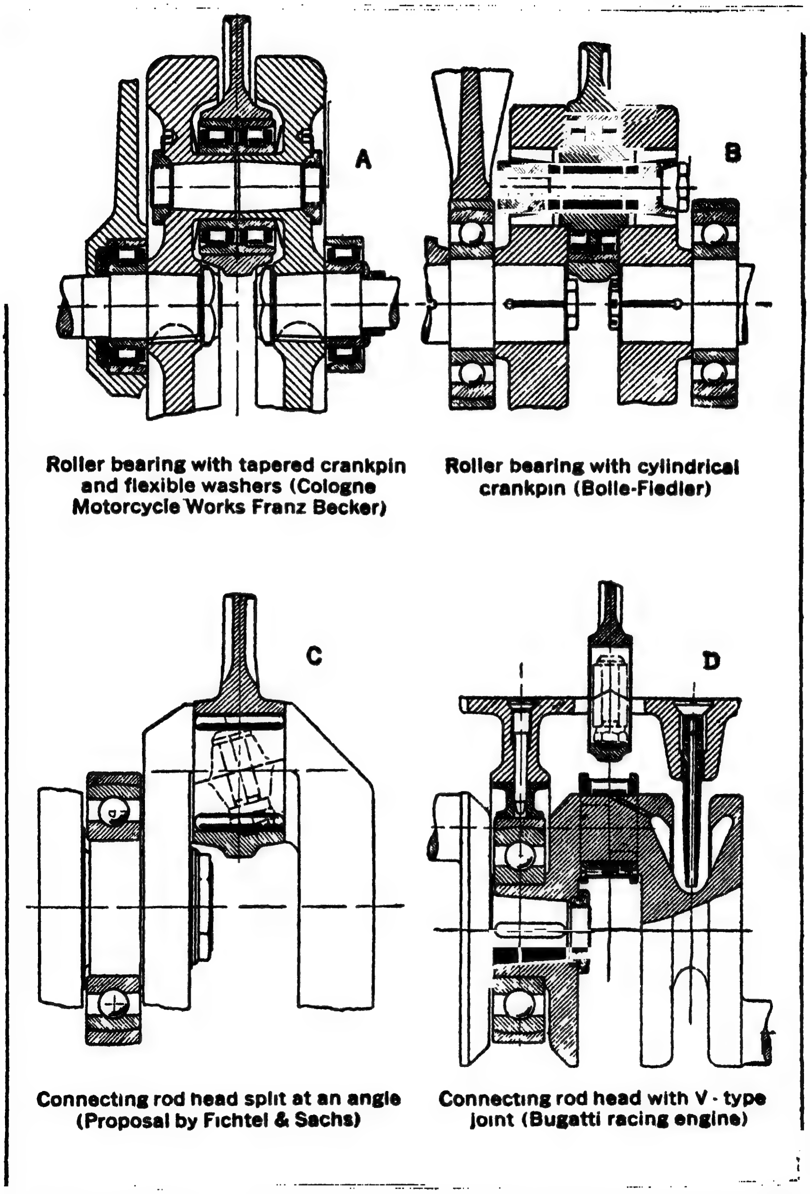


Fig. 340.—German Design Using Standard Roller Bearings in Connecting Rod Big End. B—Design in Which Crankpin Acts as Inner Race of Roller Bearing. C—German Design Showing Connecting Rod Head Split at an Angle, Using Solid Rolls. D—Connecting Rod Head with Vee Type Joint Used on Bugatti Racing Engine.

direction without hindering the tightening up of the flywheels on the crankpin. In this design, in which the inner races are mounted on extensions of the flywheel hubs, they must be made a loose fit in the first place, because drawing up of the nuts on the crankpin expands the hubs and thus results in a tight fit.

A method of construction used in automobile practice is shown in Fig. 340 B. The particular feature of this design is the method of fastening the crankpin into the crank arms. The hollow crankpin has a cylindrical fit in the arms, and its ends are cut with four saw slots and are expanded into the arms by means of tapered plugs and nuts. Keying of the joint between the arms and the pin was found unnecessary. A similar joint is used be-

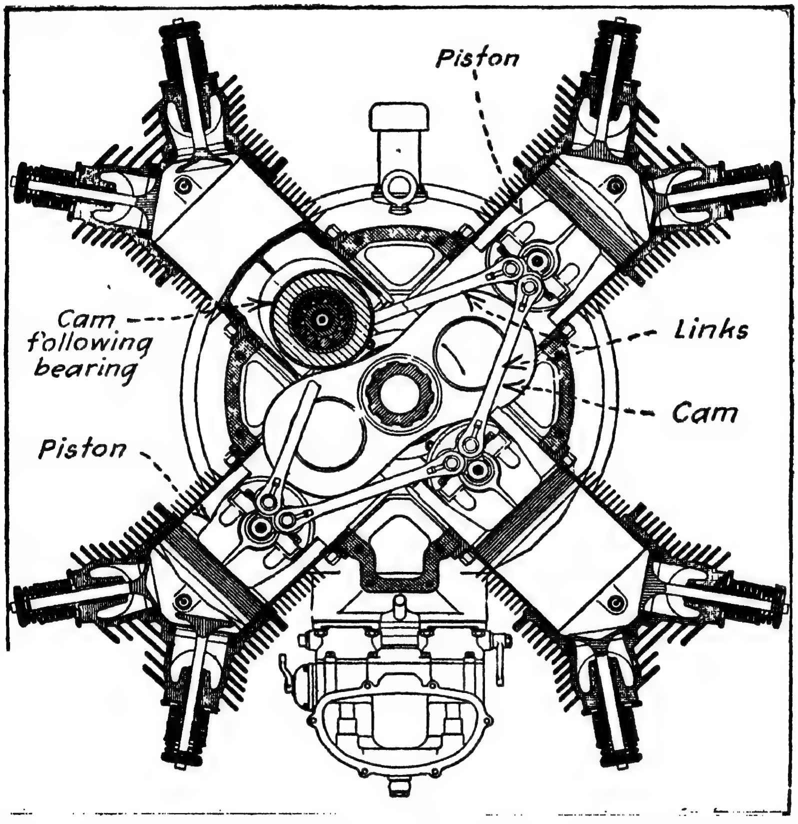


Fig. 340E.—Plan Section Drawing of Caminez Engine that Uses Cam Instead of Crankshaft for Operating Pistons. Note Use of Roller Bearing as Cam Follower.

tween the arms and the main journals, but in this case keys are used. An advantage of this construction is that it is possible to keep the longitudinal dimensions of the crankshaft accurate, which is not possible if the ordinary tapered joints are used. The construction is a very expensive one, however, even though it makes the use of anti-friction bearings possible at all points.

In the case of multi-cylinder automobile and aviation engines there is naturally a great advantage in being able to make the whole crankshaft in

a single forging, and attempts to make possible the use of roller bearings in the connecting rod heads of such cranks have not been wanting. Thus Robert Conrad, in 1903, developed a design of crankshaft in which the crank arms were of round section and substantially the same diameter as the crankpins, so that the unsplit connecting rod head and the cage could be stripped over the crank from one end, the rollers evidently being filled in by the eccentric method. Difficulties arose in connection with the case-hardening of the bearing surfaces and the assembling of the connecting rods with the crankshaft.

Split Connecting Rod Big Ends.—To facilitate assembly it is necessary to split the connecting rod head, and in order to obviate difficulties due to discontinuity of the bearing surface for the rollers, in the design shown in Fig. 340 C the head is split at an angle. One objection to this construction is that the connecting rod bolts are working under rather unfavorable conditions; another, that the slightest axial movement between rod and cap throws the bearing surface out of round. An improvement on this design is shown in Fig. 340 D, where the joint between the rod and the cap is Vee-shaped. Here the cap is held rigidly in position against endwise movement. Of course, in all of these constructions the rod and cap must be ground out together, and it is probably also the best plan to harden both parts while bolted together. In spite of these various precautions it is still very difficult to secure an accurate round bore in a two-part connecting rod head. The joint, moreover, still presents an unsolved problem and while such a joint will give results of varying value with rollers, it cannot be used at all if ball-bearings are employed and steel balls must always run in tracks or races without joints. When either of the constructions shown at C or D are used, the use of rollers is imperative.

Rollers Used as Cam Follower in Unusual Engine.—An unconventional method of power application is shown at Fig. 340 E, where a novel use is made of roller bearings which are of standard size and manufacture.

In the Fairchild-Caminez engine, the four cylinders are arranged radially about a central rotatable cam as shown in Fig. 340 E. This cam is of the double lobed type, shaped generally like a figure 8. A roller bearing is mounted in each piston, the outer race of which acts directly upon this drive cam. Adjacent pistons are connected by a system of links, the contour of the drive cam being so designed that these links maintain the piston rollers in continual contact with the cam.

The Model 447-B engine is a four-cylinder radial engine of the reciprocating piston type operating on the four-stroke cycle. The engine employs the Fairchild-Caminez drive cam mechanism in which reciprocating motion of the pistons is converted into rotary motion of the propeller shaft by means of rollers in the piston operating on a double lobed cam. The mechanism is such that each piston completes four strokes per revolution of the propeller shaft. With the four stroke cycle that is used, each piston, therefore, completes a power stroke every revolution of the shaft. It is due to this that a high power output is obtained per cubic inch of piston displacement at a low propeller speed, the shaft speed in this engine being one-half that of a crank engine of equal piston displacement for the same power output.

The main shaft of the engine is a straight alloy steel shaft to which the drive cam is splined. This shaft is supported in the engine case at the rear end by a roller bearing. The front main shaft bearing is a deep groove radial ball bearing which takes all the thrust load on the shaft and part of the axial load. The center plain bearing on this shaft is fitted with large clearance so that it takes but little of the axial load and acts mainly as a means of transmitting the lubricating oil from the case to the shaft, from where it is distributed throughout the engine.

QUESTIONS FOR REVIEW

1. Why are aluminum pistons better than cast iron for aircraft motors?
2. What is the simplest method of wristpin retention?
3. What is a strut type piston; a slipper piston?
4. Outline temperatures at various parts of a piston.
5. What is the property of aluminum that makes it suitable for piston use despite its low melting point?
6. What does the slot in the skirt of a piston do?
7. What causes piston slap?
8. Why is the allowance for expansion different in alloy pistons than in cast iron and how does it differ?
9. Why are piston rings used? Compare the two main types.
10. What is the best material for piston rings and why?
11. Describe various forms of compound piston rings and state why they are not generally used in aircraft engines.
12. How are connecting rods made for V and W engines?
13. Describe connecting rod assembly of typical static radial engine.
14. What are the disadvantages of anti-friction bearing big ends in connecting rods?
15. What is the best material for connecting rods?

CHAPTER XXIII

CRANKSHAFT AND CRANKCASE CONSTRUCTION

Influence of Cylinder Number on Crankshaft Design—Two Cylinder Engines—Four Cylinder Engines—Six Cylinder Aviation Engines—Antivibration Devices—The Ricardo Device—Aircraft and Auto Engines Different—Crankshaft Construction—Counterbalanced Crankshafts—Antifriction Bearing Crankshafts—Radial Engine Crankshafts—Securing Engine Balance Important—How Engine Parts Are Balanced—Firing Balance Important—Normalized Steel—Camshaft Influence on Crankcase Design—Engine Base Construction—Special Requirements Dictate Crankcase Design—Radial Engine Crankcases—Packard X Engine Crankcase—Antifriction Bearings—Lightness of Construction—Materials Used in Engines—Properties of Aluminum Alloys—Cylinder and Crankcase Retention Bolts—Alloy Steel Bolts—Heat Treated Carbon Steel Bolts—Hardness Testing Methods—The Brinell Test—The Scleroscope Method.

Influence of Cylinder Number on Crankshaft Design.—The number and location, or rather arrangement of cylinders has a material influence on crankshaft and crankcase design, the designer having no choice about the type or length of crankshaft to be used once he determines the number of cylinders he will use and how he intends to place them. When he designs a four- or six-cylinder engine of the conventional in-line form he can be guided largely by previous automobile practice but when he starts to lay out an engine of eight or more cylinders, a variety of cylinder arrangements present themselves. An eight-cylinder engine may be a twin four in Vee arrangement and a twelve-cylinder may be a twin six in Vee arrangement or a W type with three banks of four cylinders each. Engines with an odd number of cylinders must be radial engines with the possible exception of the three-cylinder which can also be an in-line form, but five, seven and nine cylinders must have a radial arrangement to secure an even firing order. The reasons why engines of varying number of cylinders are used should be given consideration before the influence of cylinder number on crankcase and crankshaft design is considered. We will consider the various possible conventional cylinder arrangements in order.

Two-Cylinder Engines.—The simplest form of motive power suitable for aircraft use is the two-cylinder horizontal-opposed engine. From the standpoint of mechanical balance this engine was greatly superior to the twin-cylinder Vee-type, used in motorcycles, as will appear by reference to Fig. 341 B, where the inertia forces due to the reciprocating parts have been graphically shown, the forces acting on one piston being at all times offset by an equal and opposite force acting on the other piston. However, there is still a couple about the center of the engine. which interferes with perfectly smooth running. This couple is shown in Fig. 341 A, and is the result of the offset relationship between the two cylinders. When used in automobiles the individual power impulses at the lower speeds were still uncomfortably apparent in larger sizes and in the matter of flexibility there was much to be desired. Lubrication and carburetion problems also were responsible for abandoning the two-cylinder horizontal engine for the four-

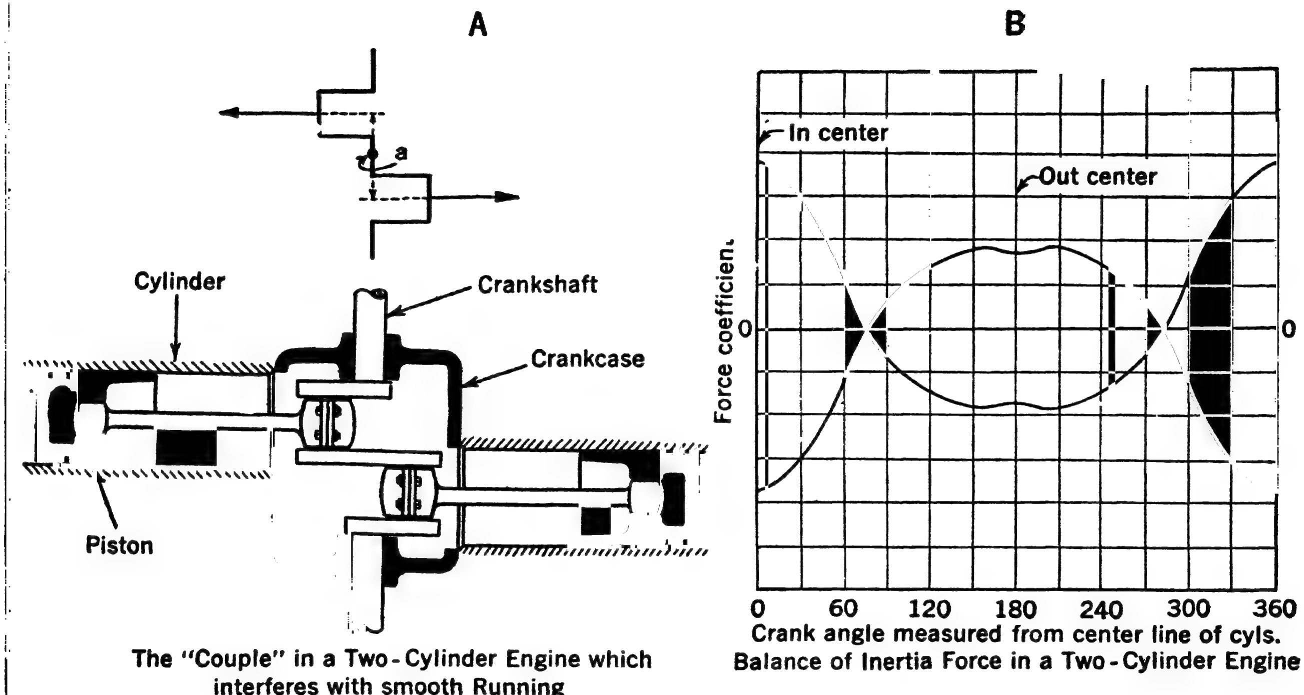


Fig. 341.—Diagram at A Showing the Couple which Interferes with Smooth Running in an Opposed Two-Cylinder Engine. Diagram At B Shows Balance of Inertia Force in a Two-Cylinder Engine.

cylinder type. Furthermore, with a fixed chassis width the two-cylinder opposed engine could not meet the demand for more power and consequently larger engines in automobiles. The same holds true in airplanes, the fuselage width limiting to some extent the size of a two-cylinder engine that can be conveniently installed. Low-powered, high-speed aviation engines of the two-cylinder opposed type are operative but about twenty horsepower per cylinder seems to be the practical limit beyond which it is not desirable to go.

Four-Cylinder Engines.—A three-cylinder engine was featured in two-cycle engines and in at least one case in a four-cycle engine, but it was soon realized that the four-cylinder vertical four-cycle engine had many advantages, the chief of which can be summarized as follows: The impulse frequency is fairly satisfactory, there being two explosions per revolution. The engine is compact and accessible and can be built in units giving rather high horsepower, the crankshaft is sturdy and easily manufactured and the problem of distributing the carburetted mixture is particularly easy to solve. This is due to the symmetrical arrangement of the inlet manifold that is possible owing to the absence of overlapping suction impulses. The limiting factor is vibration due to the inertia forces not being cancelled out as in the six-cylinder engine. This can best be understood by reference to Fig. 342 A, which shows a condition common to all four-cylinder engines with cranks at 180 degrees. When one pair of pistons is in mid-stroke position, the other pair occupies a lower position, the extent of this difference depending upon the angularity of the connecting rod. Since the inertia forces acting on the pistons vary with different positions in the stroke, it is clear that the forces acting on the pair of pistons in mid-stroke are not equal and opposite to those acting on the other pair. The diagram of the inertia forces in a four-cylinder engine, prepared by Col. J. G. Vincent, shown in Fig. 342 B clearly indicates the relative value of these forces at different crank angles and the extent to which they cancel each other. The resultant force shown is, of course, obtained by combining the two curves and represents the net out-of-balance force which, acting on the engine as a unit, produces vibrations having a periodicity coincident with the several power impulses.

At certain engine speeds a synchronous vibration is produced which results in a disagreeable sensation. This is, of course, more pronounced in the case of large four-cylinder engines in which these vibrations occur and are said to coincide with the "period" of the engine and the cause of these periods can be readily understood by a simple analogy. It is well known that a very slight push given at just the right time will keep a swing or pendulum oscillating, whereas the same force applied at certain other times will tend to stop it. When this situation was realized, attempts were made to minimize the out-of-balance condition. Reciprocating parts were lightened, as by using aluminum pistons and dural rods, and connecting rods were lengthened, it being understood that decreased angularity of the connecting rod, as shown in Fig. 342 A, tends to bring about better balance conditions. Lanchester, in England, must be credited with some very ingenious work which had for its object mechanism which would generate forces at the right time to offset the unbalanced forces in a four-

cylinder engine, and which is shown at Fig. 344, and, had it not been for the demand for more flexible and smoother engines, there is little doubt that the four-cylinder out-of-balance conditions would have been overcome in a practical way by such mechanism.

Six-Cylinder Aviation Engines.—The next step forward was the six-cylinder engine and it immediately answered the demand for more power, more flexibility, more frequent power impulses and greater smoothness,

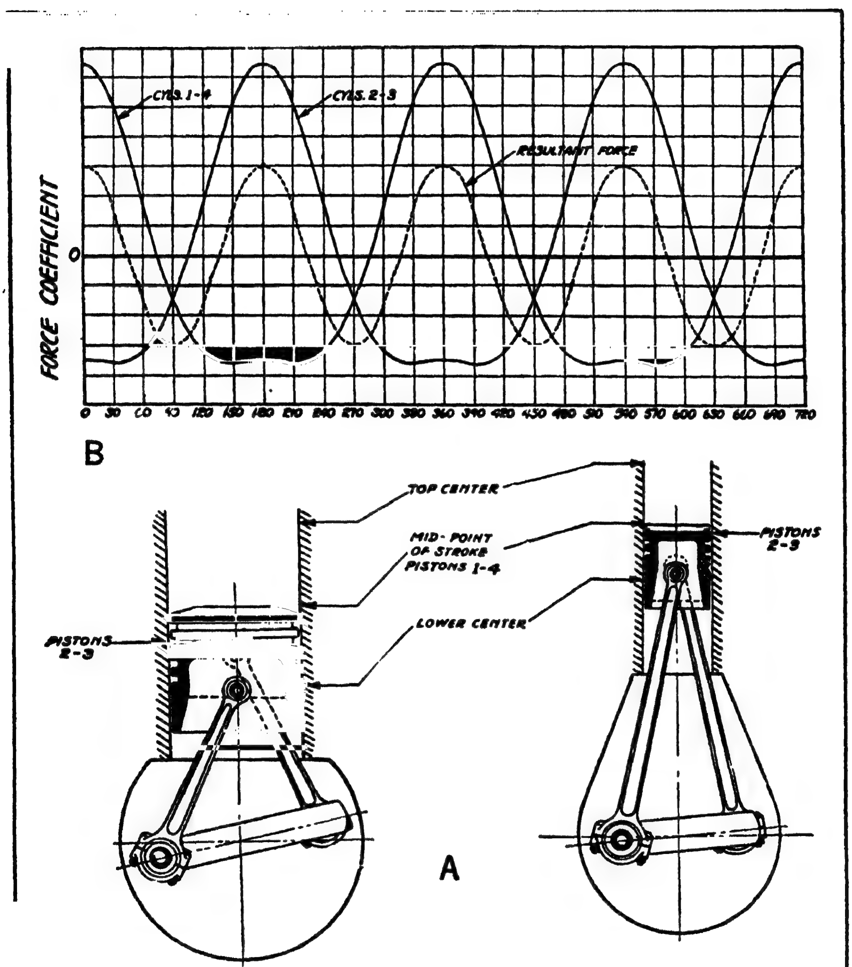


Fig. 342.—Diagrams Showing the Cause of Unbalanced Forces in a Four-Cylinder Engine. Comparison Between Short and Long Stroke Types Shown at A. B Shows Inertia Force in a Four-Cylinder Engine.

which is primarily due to the inherent balance of the rotating and reciprocating parts. The diagram of inertia forces in a six-cylinder engine, shown in Fig. 343, clearly indicates the relative value of these forces at different crank angles and how they cancel out, due to the location of the crankpins

at 120 degrees with relation to each other. The resultant force is, of course, obtained by combining the three curves and graphically represents the perfect balance obtained by this arrangement. The six-cylinder engine is undoubtedly a type that will be represented for years to come in automobiles, owing to its inherent advantages; however, there are certain disadvantages which may be termed constructional that set definite limitations in its design, and its application to aircraft. The crankshaft of a six-cylinder engine is necessarily long and space and weight limitations do not permit of the

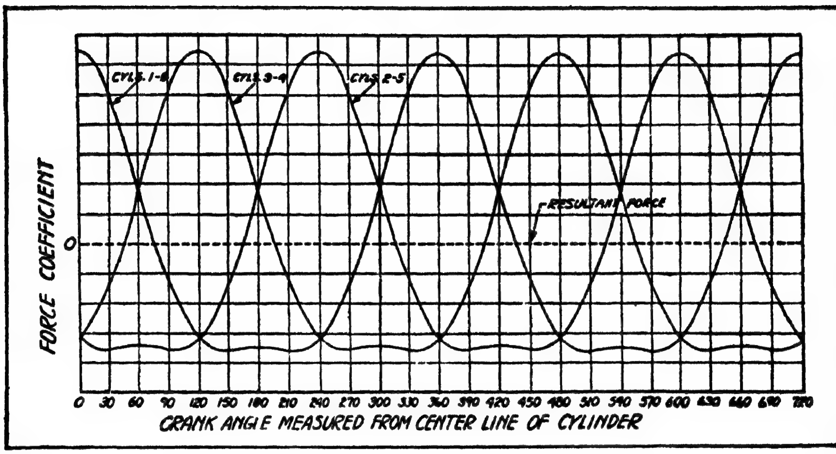


Fig. 343.—Diagram of Inertia Force in Six-Cylinder Engine Showing Superior Balance Obtained by the Use of that Number of Cylinders.

shaft being made as rigid as could be desired. In actual practice, therefore, there is a certain amount of periodic twisting of the crankshaft which is compensated for in automobile engines by various systems of counterbalances or by using vibration dampeners of various types. In aviation engines, the materials and proportions of the shaft are such that the twisting is reduced to a point where it is not objectionable in the power range in which six-cylinder engines are usually built.

Anti-Vibration Devices.—This vibration aspect of four-cylinder engine design would more properly be termed improvement in means for reducing engine vibration, since vibration is by no means eliminated when complete balance is obtained, a fact well known to most designers of six-cylinder and even twelve-cylinder engines, so vibration is not only found in the simpler types. In respect to engine balance *per se*, the position to date is that engines with two, four, six, eight or twelve cylinders, are or can be completely balanced in respect to primary and secondary unbalanced forces and couples. The six- and twelve-cylinder engines are of course inherently balanced; the four- and eight-cylinder need the application of a device for neutralizing the secondary unbalanced forces. Fig. 344 shows the Lanchester anti-vibrator, and Fig. 345 the Ricardo secondary balancing device, each of which is effective in automobile engines. The principle of the Lanchester device is that of two reverse-rotating bob-weights that apply equal and opposite forces to the secondary inertia forces set up by pistons at the

end of each stroke, the bob-weights rotating at twice the engine speed. As the energy content of the bob-weight system is constant at a constant speed, the only force required to drive the device is that arising from the friction of the bob-weight spindles, so that the driving mechanism is merely a motion transmitter. When the engine is accelerated or decelerated, tooth pressures of material magnitude arise, but can be dealt with easily in the

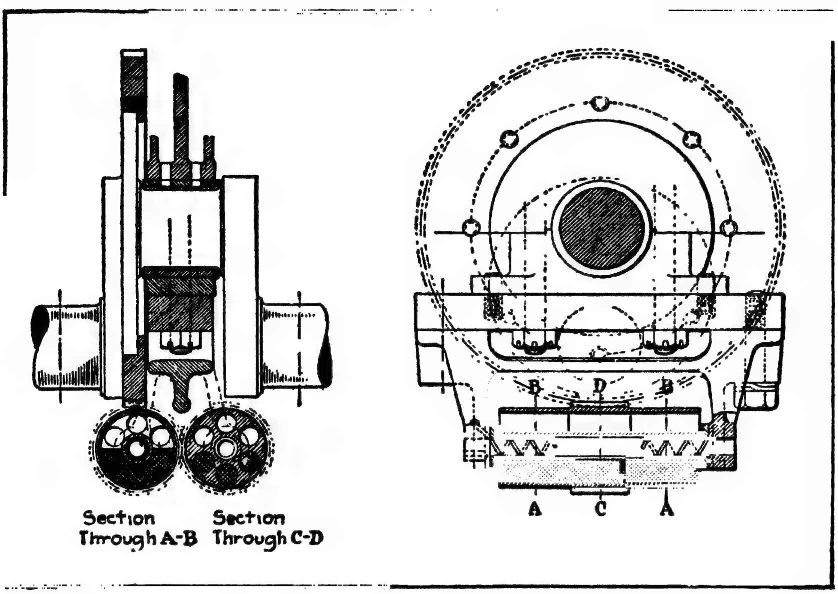


Fig. 344.—Diagram Showing Construction of Lanchester Anti-Vibrating Device for Four-Cylinder Engines.

design of the driving gearing. The peripheral speeds of the gears is 90 feet per second at an engine speed of 3,000 r.p.m., approximately the same as with many turbine reduction-gears in daily use, where, in addition, the tooth pressures are exceedingly high.

The Ricardo Device.—The Ricardo device shown at Fig. 345 is based upon the principle of introducing reciprocating masses driven by linkages, producing the same angular effects in respect to inertia with these masses as the connecting rod crank system produces with the pistons. It is exceedingly ingenious. The conditions under which the pin-joints in the linkage systems work are no worse than those of the piston-pin, and although the device appears a little complicated, it is simple in detail and works well. It has not, however, the advantage of ready application to existing designs possessed by the Lanchester anti-vibrator. Whatever effect counterbalancing a crankshaft may have in reducing bearing loads, it certainly does not affect the balancing of secondary forces one way or the other. The only virtue of crankshaft counterbalancing, the addition of balance weights to the crank webs so that each individual crank and attached rotating masses may be balanced, is that wear on the main-bearings of the crankcase can be greatly reduced and the crankcase itself reduced in weight very considerably because fewer main-bearings are needed.

The whipping tendency or "skipping-rope action" of an unbalanced crankshaft, particularly in a six-cylinder engine, sets up very heavy bearing loads and crankcase stresses, so that counterbalancing may be very desirable in engines with center crankshaft bearings. The use of counterbalance-weights should be accompanied by an increase in crankshaft diameter, owing to the torsional mass effects of the balance-weights.

As any expedient which increases weight without producing a useful return in power is undesirable in airplane engines, counterbalances and anti-vibration devices are seldom used in airplane engines except in the static radial forms where a number of connecting rods are joined to one crankpin.

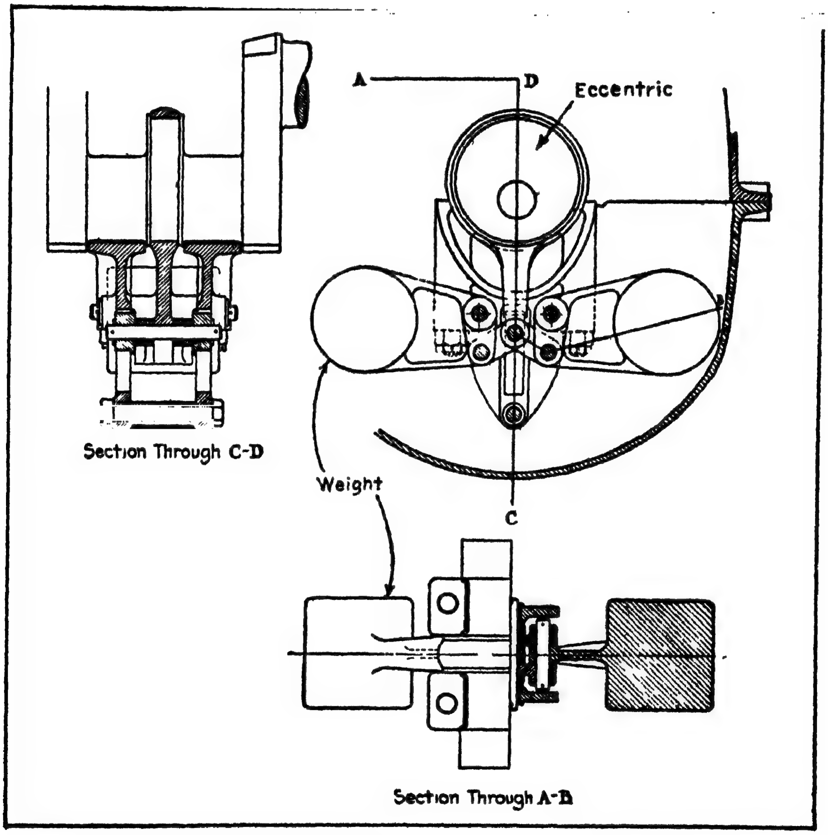


Fig. 345.—The Ricardo Four-Cylinder Engine Balancing Mechanism.

Aircraft and Auto Engines Different.—When we consider aircraft engines in relation to the desirable number of cylinders and their Vee arrangement we are faced with altogether different problems. Although the aircraft engine owes its origin to the passenger-car engine, there is little in common between the two types. For instances, low weight is a prime consideration for aircraft work; in a car it is of secondary importance. The radial cylinder air-cooled engine that is so widely used in airplanes is a

type that could not be used in motor cars because of difficulties met with in installation and cooling. Noisy operation is not a handicap with the aeronautic engine, since the propeller noise will always be a bar to silent operation of the powerplant, whereas quiet operation of the passenger-car engine is one of the points most sought. Good fuel economy under full load is a prime requisite for the aeronautic engine, in the passenger-car engine this is a matter of far less importance, good performance and flexibility with adequate gasoline mileage on partial loads being the factors striven for. There is one point in common between airplane and passenger-car engines, which has a determining influence in the number of cylinders, and that is vibration. In the passenger car we seek for a smooth engine because that is what the passenger appreciates, in the airplane we must have a smooth engine, since neither the engine nor the structure of the plane can withstand the vibrations produced by a rough engine.

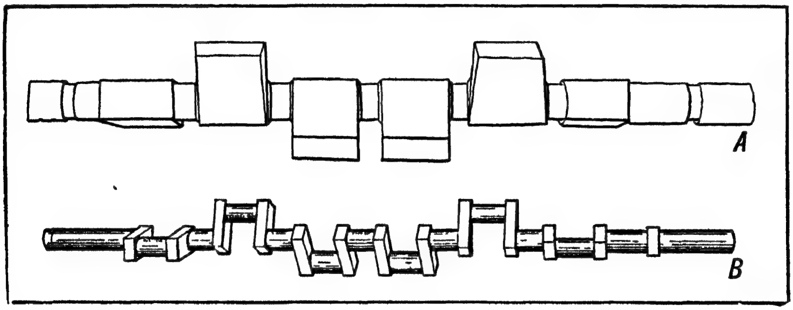


Fig. 346.—Showing Method of Machining Crankshaft from Machine Forging. A—Appearance of the Rough Steel Forging Before Forming. B—The Finish Six Throw Seven Bearing Crankshaft Made from the Forging Shown at A.

The airplane engine is mounted on a few relatively frail sticks or tubes compared to the automobile frame strength or on some light metal stampings that form an adequate support so long as they are subjected to reasonable stresses, but the structure of an airplane cannot long endure the fatiguing stresses of a rough engine. The four-cylinder engine in its larger sizes was therefore never popular for aircraft use. The six-cylinder engine we have seen is inherently in perfect running balance, but its length makes it rather heavy for a given power. Simplicity of manufacture, ease of installation, good accessibility and low head-resistance, contributed to making the eight- and twelve-cylinder Vee types ideal where high-powered water-cooled types are needed, and the seven- and nine-cylinder radial engines are not to be lightly considered where accessibility and compactness combined with simple installation procedure and air cooling are demanded. On account of the desirability of keeping aviation engine weight to the minimum, it is doubtful if secondary force balancing mechanisms such as designed by Lanchester or Ricardo would be as good as using more cylinders where added weight produces useful power. The same thing applies to crankshaft counterbalances when more than one crankpin is employed.

Crankshaft Construction.—The importance of the crankshaft has been previously considered, and some of its forms have been shown in views of

the motors presented in earlier portions of this work. The crankshaft is one of the parts subjected to the greatest strain and extreme care is needed in its construction and design, because the entire duty of transmitting the power generated by the motor to the airscrew devolves upon it. Crankshafts are usually made of high-tensile strength alloy steel of special composition. They may be made in four ways, the most common being from a drop or machine forging which is formed approximately to the shape of the finished shaft and in very rare instances (experimental motors only) they may be electric steel castings. When only a small number of crankshafts are needed they are made from machine forgings, which call for considerably more machine work than would be the case where the shaft is formed between dies, a procedure only possible where production warrants their cost. Some engineers favor blocking the shaft out of a solid

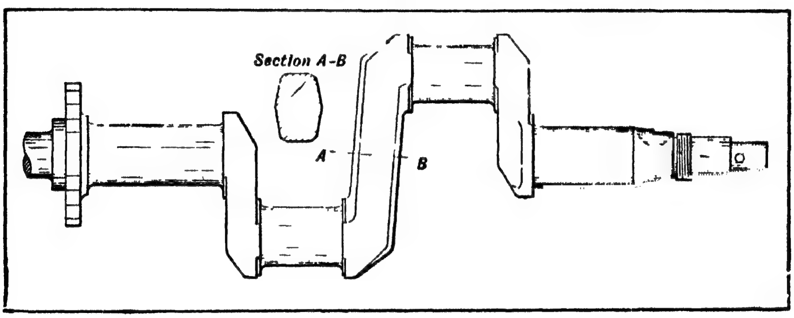


Fig. 347.—Showing Design of Crankshaft for Twin-Cylinder Opposed Motor. Web Sections Must be Greatly Lightened and Main Journals and Crankpins Should be Bored Out to Reduce the Weight, if Used in an Aviation Engine.

slab of metal and then machining this rough blank to form. In some radial-cylinder motors of the Wasp, Ryan-Siemans, Bristol-Jupiter and other types, the crankshafts are built up of two pieces, held together by taper and key fastenings or bolts as will be described more in detail later.

The form of the shaft depends on the number of cylinders and the form has material influence on the method of construction. For instance, a four-cylinder crankshaft could be made by either of the methods outlined. On the other hand, a three- or six-cylinder shaft is best made by the machine forging process, because if drop forged or cut from the blank it may have to be heated and the crank throws bent around so that the pins will lie in three planes 120 degrees apart, while the other types described need no further attention, as the crankpins lie in planes 180 degrees apart. This can be better understood by referring to Fig. 346 showing a shaft in the rough and finished stages. At A the appearance of the machine forging before any of the material is removed is shown, while at B the appearance of the finished crankshaft is clearly depicted. The built-up crankshaft is seldom used on multiple-cylinder motors, except in some cases where the crankshafts revolve on ball-bearings as in some racing engines. The general design of the crankshaft depends upon the number of main bearings to be used, and this in turn may depend upon the whim of the designer or on some particular feature of cylinder arrangement which necessitates a certain arrange-

ment of main bearings. In general, a designer tries to make an engine as short as possible, and the number of main bearings may have a considerable influence on the length of the engine. From a rigidity standpoint a bearing on each side of each crankthrow is desirable, and this has become practically universal practice in aircraft engines. This construction also simplifies the drilling of the crankshaft for oil distribution to the crankpins.

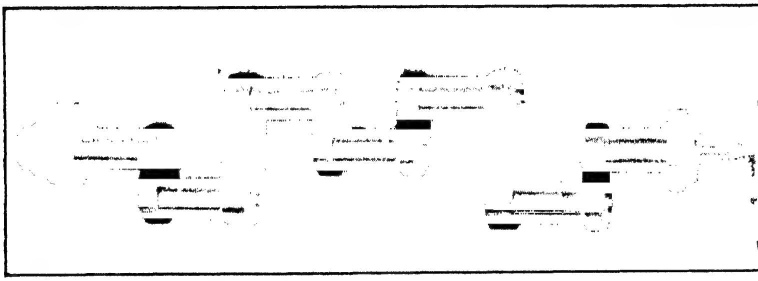


Fig. 348.—Crankshaft of an Eight-Cylinder "Vee" Engine for Aviation Use that Employed Side by Side Connecting Rod Big Ends. Note Length of Crankpin Bearings Necessary to Accommodate Two Rods.

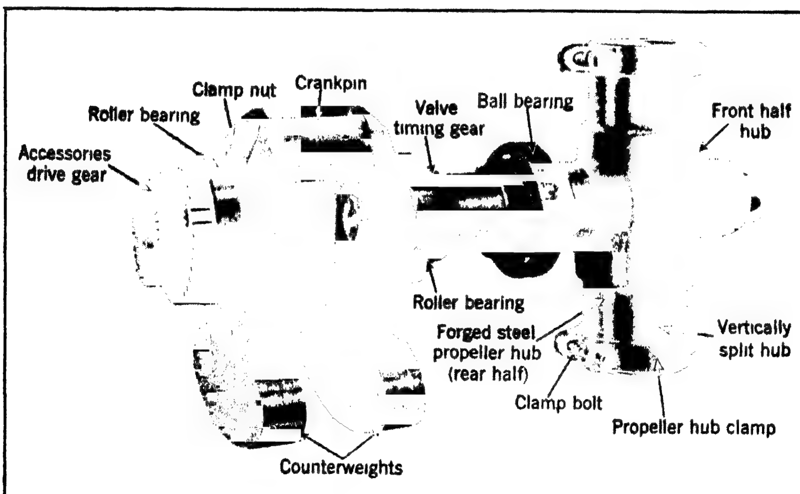


Fig. 349.—Crankshaft Assembly of Pratt & Whitney "Wasp" Motor is Built-Up of Two Members but Becomes One When Bolted Together. Note the Method of Mounting Anti-Friction Ball Bearings, and the Split Propeller Hub of a Standard Steel Propeller.

Crankshaft form will vary with the number of cylinders and it is possible to use a number of different arrangements of crankpins and bearings for the same number of cylinders. The simplest form of crankshaft is that used on radial cylinder motors, as it would consist of but one crankpin, two counterbalanced webs, and the crankshaft, as shown at Fig. 349. As the number of cylinders increase in Vee and in-line motors, as a general rule, more crankpins are used. The crankshaft that would be used on a two-cylinder opposed motor is shown at Fig. 347. This has two throws and the

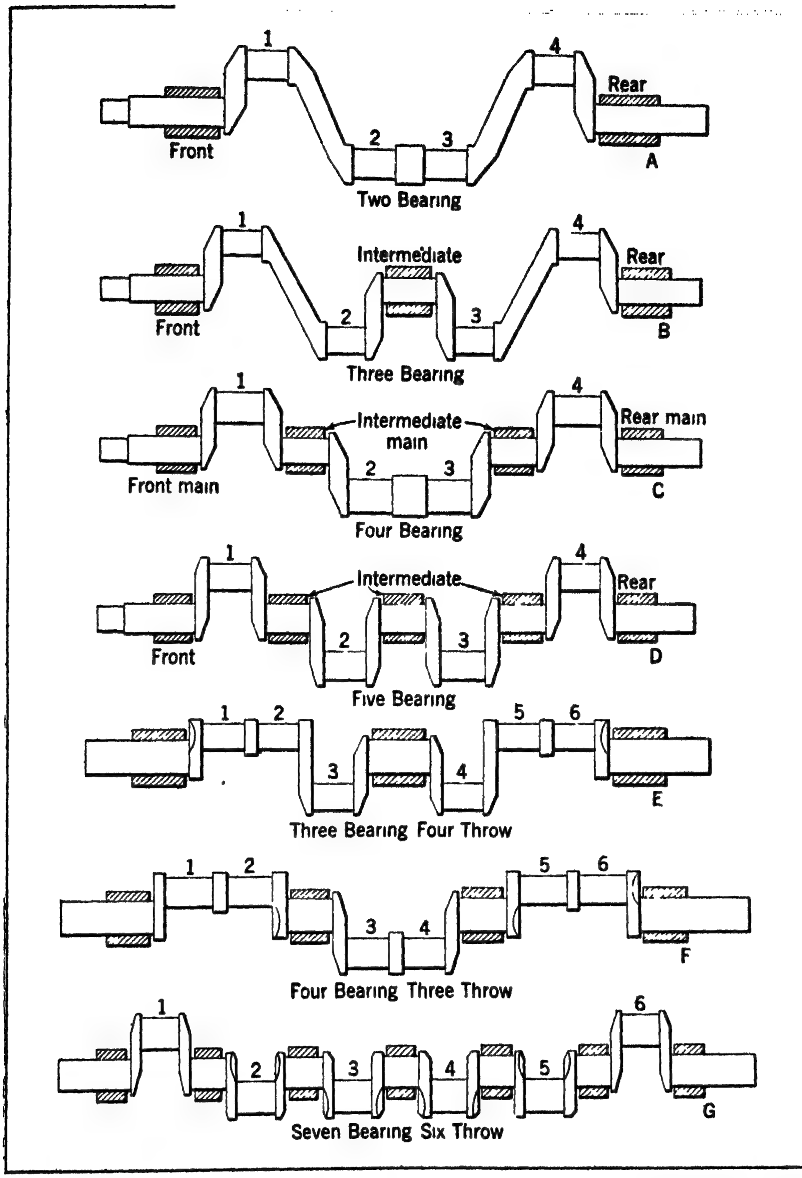


Fig. 350.—Various Types of Four and Six Throw Automotive Crankshafts, Showing Possible Combinations of Main Bearings.

crankpins are spaced 180 degrees apart. The bearings are exceptionally long. Four-cylinder crankshafts may have two, three or five main bearings and three or four crankpins as shown at Fig. 350 A, B, C and D. In some forms of two-bearing crankshafts, such as used when four cylinders are cast in a block, or unit casting, two of the pistons are attached to one common crankpin, so that in reality the crankshaft has but three crankpins

as shown at Fig. 350 A. A typical three bearing, four-cylinder crankshaft is shown at Fig. 350 B. The same type can be used for an eight-cylinder Vee engine, except for the greater length of crankpins to permit of side by side rods, as in the crankshaft shown at Fig. 348. Six-cylinder vertical tandem and twelve-cylinder Vee engine crankshafts usually have four or seven main bearings depending upon the disposition of the crankpins and arrangement of cylinders. At Fig. 351 A the bottom view of a twelve-cylinder engine with bottom half of crankcase removed is given. This illustrates clearly the arrangement of main bearings when the crankshaft is supported

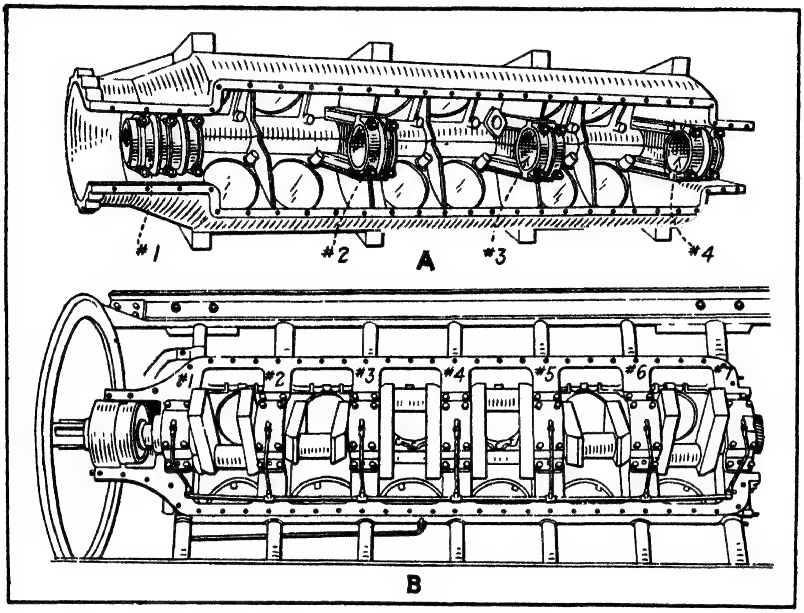


Fig. 351.—Crankcase and Crankshaft Construction of Early Twelve-Cylinder Motors. A—The Deusenberg Crankcase Had Four Main Bearings. B—The Curtiss Crankshaft Was Supported by Seven Main Bearings.

on four journals. The crankshaft shown at Fig. 351 B is a twelve-cylinder seven-bearing type. A modern crankshaft of alloy steel employed on a twelve-cylinder Vee engine is clearly shown at Fig. 353. The removable bearing liners and the main bearing caps are also shown. The drilled out main journals are closed by stamped end plates held by the bolts T so each crankpin receives lubricant from an adjoining reservoir in a main shaft journal.

Counterbalanced Crankshafts.—In some automobile engines, extremely good results have been secured in obtaining steady running with minimum vibration by counterbalancing the crankshafts as outlined at Fig. 352. The shaft at A is a type suitable for a high-speed four-cylinder vertical or an eight-cylinder Vee type. That at B is for a six-cylinder vertical or a twelve-cylinder Vee with scissors joint rods. While counterbalancing crankshafts helps in an automobile engine, its advantages are not of enough moment in airplane engines to justify the crankshaft weight increase except in radial

engines using a very short shaft, as shown at Fig. 349. As previously stated, counterbalances cannot absolutely eliminate engine vibration as they do not cancel out the forces causing it.

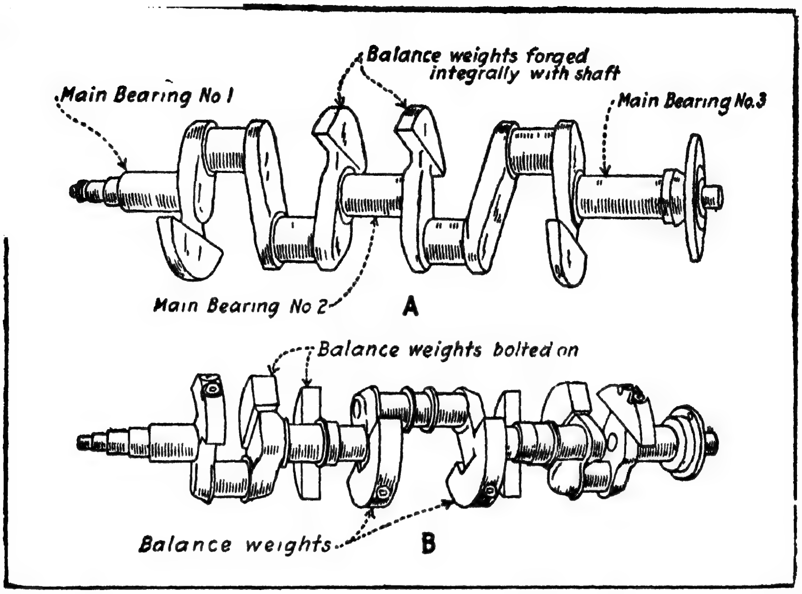


Fig. 352.—Use of Balance Weights in Automotive Engines to Counterbalance Crankshafts Reduces Engine Vibration and Permits High Rotative Speeds.

Anti-Friction Bearing Crankshafts.—While crankshafts are usually supported in plain journals there seems to be a growing tendency to use anti-friction bearings of the ball or roller type for their support. This is especially noticeable on radial motors where but two or three main bearings are utilized as shown at Fig. 349. When ball-bearings are selected with proper relation to the loading which obtains they will give very satisfactory service as often they will outwear other parts of the engine. They permit the crankshaft to turn with minimum friction, and if properly selected will never need adjustment. The front end is supported by a bearing which is clamped in such a manner that it will take a certain amount of load in a direction parallel to the axis of the shaft, while the rear end is so supported that the outer race of the bearing has a certain amount of axial freedom or "float." The inner race of each bearing should be firmly clamped against shoulders on the crankshaft. At the rear end of the crankshaft timing gear a suitable locking arrangement is used, while at the front end the bearing is clamped by a threaded retention member between the propeller hub and a shoulder on the crankshaft. The hub is held in place by a taper and key retention. The ball-bearings are sometimes carried in a light housing of bronze or steel when installed in aluminum cases, which in turn are held in the crankcase by bolts. The Hispano-Suiza engine uses ball-bearings at front and rear ends of the crankshaft, but has plain bearings around intermediate crankshaft journals. The radial engines would not be really

practical if ball or roller bearings were not used, as the bearing friction and consequent depreciation would be very high.

The crankshaft shown at Fig. 354 is a foreign design used in a twelve-cylinder Vee engine and revolves on eight anti-friction bearings, one being installed each side of each crankpin and one at the extreme front end of the shaft. The bearing inner races are sufficiently large in bore to permit

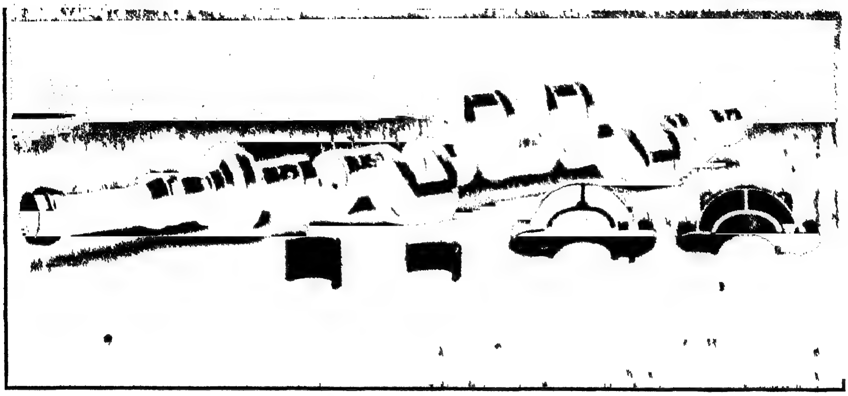


Fig. 353.—Crankshaft of Fiat A20 Aviation Engine Showing Splines for Driving Propeller and Also Method of Lightening Crankshaft by Machining Crank Webs and Boring Out all Crankpins and Main Journals. The Bearing Shells and the Main Bearing Journal Caps are Also Shown in this View.

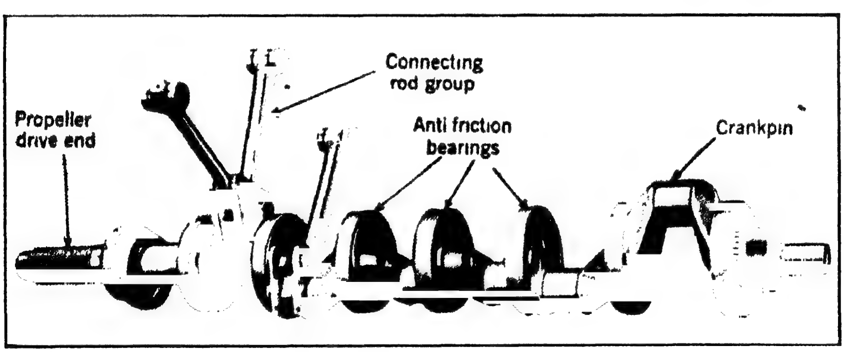


Fig. 354.—Crankshaft Assembly of Foreign Aviation Engine Showing Anti-Friction Bearings Installed on Main Journals.

passing intermediate bearings over the rounded and beveled crank webs so installation is not difficult and a one-piece shaft may be employed. The longitudinal sectional view of the Napier Lion engine at Fig. 355 shows the manner of installing roller bearings very clearly. As the engine is a twelve-cylinder W type a four-throw crankshaft suffices and five main bearings of the roller type support explosion loads. Because the engine is a geared form, other roller bearings are needed to support the reduction gear shaft and the shaft driving the air screw which turns slower than the crankshaft. The four larger diameter roller bearings on the crankshaft, which have

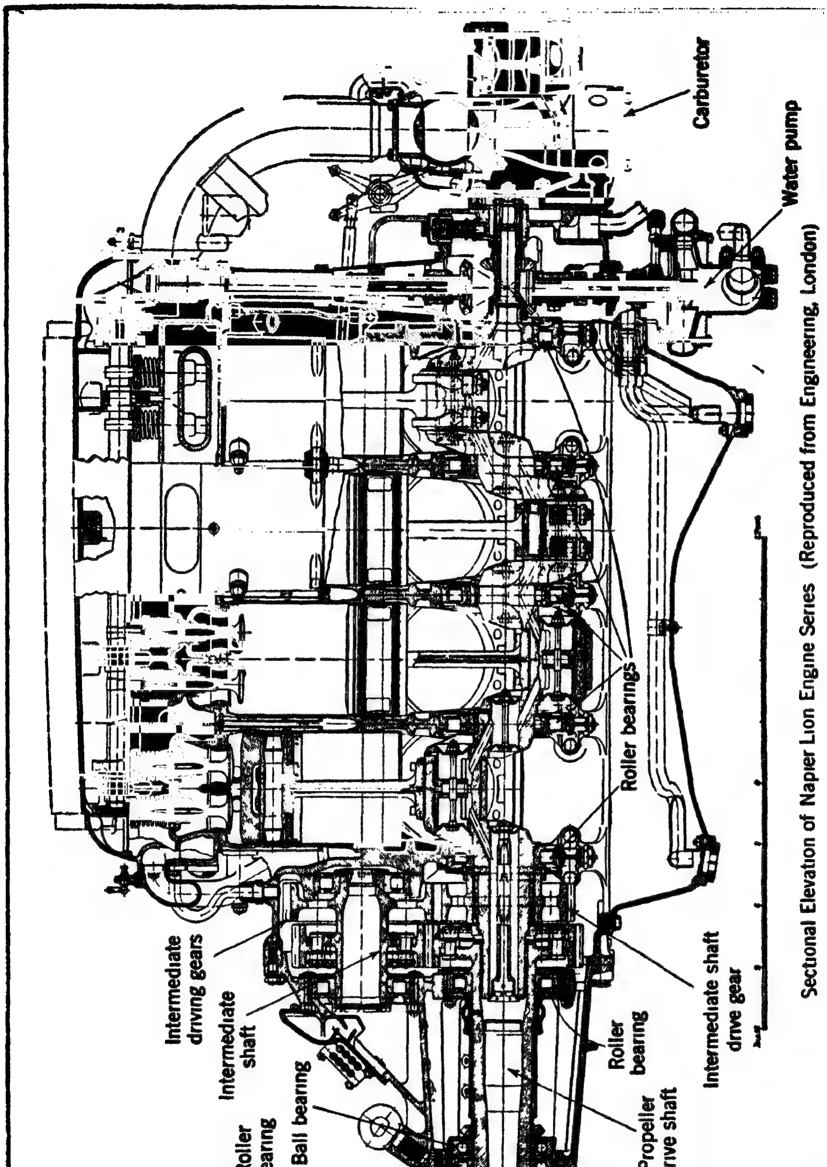


Fig. 355.—Sectional Elevation of the Napier-Lion "W" Type Aviation Engine, Showing Method of Supporting Crankshaft by Roller Bearings, Also Making Other Constructional Details Clear.

straight rolls restrained against end movement by flanged inner races have outer races which do not restrain the rolls endwise, but the smaller roller bearing at the accessories drive end restrains end movement of the rolls and consequently of the entire shaft because both inner and outer races are flanged The reason end movement of the shaft is permitted in the other four bearings is to allow for the shaft lengthening due to heat after the engine has been in operation for some time The design shown is an interesting one and is worthy of the closest study

Radial Engine Shafts.—The Wasp crankshaft assembly shown at Fig. 349 is an excellent example of American built-up crankshaft construction. The parts comprising the assembly are shown at Fig. 356. It will be observed that the crankpin, one crankweb and the propeller driveshaft are in one piece, the interior of the crankpin being provided with keyways. The other web has a short extension shaft intended to fit into the crankpin, keys fitting corresponding keyways. The two pieces are held together by a bolt passing through the crankpin into the webs. The crankshaft of the English Bristol Jupiter engine is also of the built-up type and manufactured from hardened and tempered 60 ton nickel chrome steel forgings. It is shown

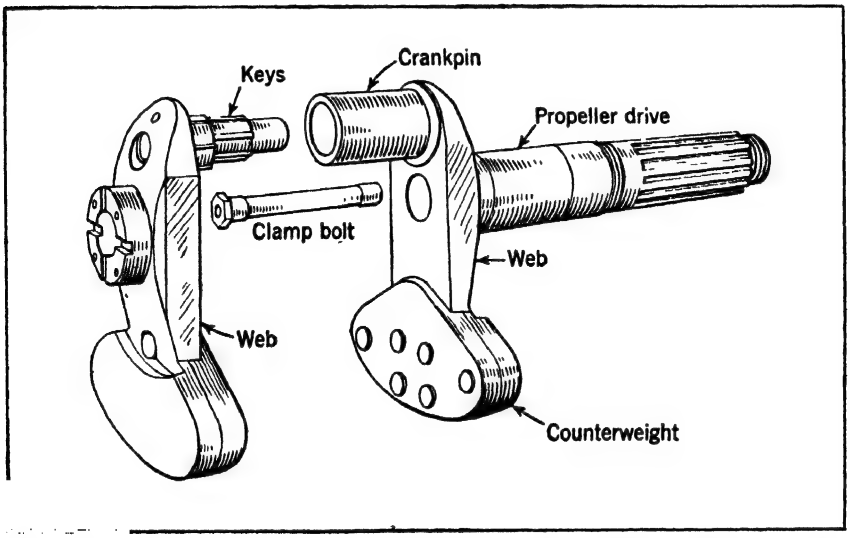


Fig. 356.—Parts Comprising the Crankshaft Assembly of Pratt & Whitney "Wasp" Motor Before they are Bolted Together. Notice the Rugged Design of the Crankpin Joint.

at Fig. 357. The crankshaft spigots in the maneton, and is registered by a stout taper key formed integral with the eye of the maneton, which is split on the crankpin center line and clamped by a large diameter bolt and nut. The joint obtained is entirely free from any trace of play or working, even under full throttle excess speed conditions. The crank cheeks have been slotted out and the balance masses pushed out in order to obtain the maximum effect for the minimum gross weight. The complete shaft is carried on two main roller bearings, located immediately behind each crank web (not shown in illustration) with a special Skefko double purpose spherical roller bearing at the propeller end, and a small white metal steady bearing at the tail end. The shaft is drilled throughout for lightness, communicating holes and blanking plugs allowing of the resulting chambers being utilized for oil circulation and distribution.

Securing Engine Balance Important.—Considerable progress has been made by machine-tool builders in providing equipment for locating and cor-

recting the unbalance of rotating parts, and the problem of selecting the machine best suited for the particular part to be balanced requires much study and investigation on the part of the engineers who specify the purchase of manufacturing equipment. L. L. Roberts, M.S.A.E., in a paper read before the Detroit Section of the S. A. E. described the processes that had been tried at the Packard factory. The crankshafts of both types of engine (automobile and aircraft) are machined all over. Of course, if it were possible to hold absolutely to certain dimensions, it would be unnecessary to make any balancing corrections.

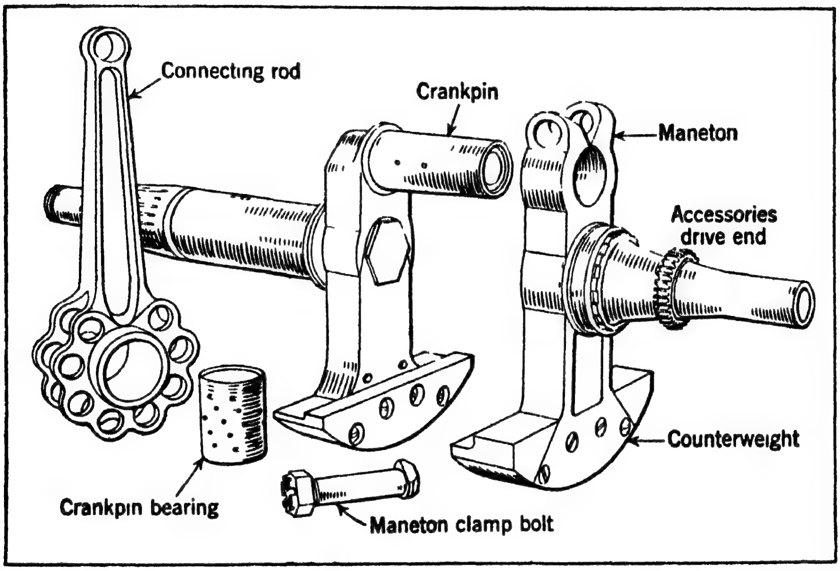


Fig. 357.—Master Connecting Rod and Crankshaft Parts of the Bristol Jupiter Nine-Cylinder Radial Air-Cooled Engine.

After checking up about 100 crankshafts carefully it was found that, from a manufacturing viewpoint, certain so-called unimportant dimensions such as the thickness and the contour of the crank-arms would cause the crankshaft to be considerably out of balance if machined to the extreme limits allowed either way of ± 0.010 inch. The first step was to determine a suitable method for correcting the existing unbalance and, with this in mind, several plants that had installed balancing machines were visited. It was found that the corrections could be made either by drilling holes in the crank-arms or by planing off stock from the corners of the crank-arms after the crankshaft had otherwise been machined completely. Both the six-cylinder and the eight-cylinder types of crankshaft are difficult to balance because of the restrictions placed on the removal of the corrective masses. A crankshaft corrected by drilling was submitted to and was rejected immediately by the Packard engineering department, because the holes tended to weaken the structure. Drilling disfigures such a shaft and gives the impression that it has been resorted to as an expedient. Further study showed that five possible methods of effecting balance remained.

These are

(1) Varying the bore of the hole in the crankpin
(2) Using calibrated plugs in the crankpin bores
(3) Beveling the crank-arm opposite the end of the crankpins
(4) Removing uniform layers from the periphery of the crank-arm
(5) Removing uniform layers of metal from the flat side of the crank-arm.

The first method was rejected because of its great expense and inconvenience. While in a certain sense it is convenient, the second method is subject to errors of fitting wrong plugs. The third method, that of beveling off the corners of the crank-arms, was tried temporarily and worked well; but this process involves considerable skill and many trials on the part of the operator and, while this method was allowed, it was a slow and costly process in production. The fourth and fifth methods seemed the most promising and were, therefore, given further study.

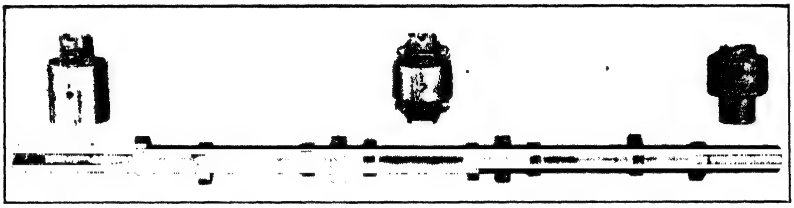


Fig. 358.—Typical Automotive Engine Camshaft with Valve Lifting Cams Forged Integrally.

How Engine Parts are Balanced.—A precision type of balancing machine, in which the crankshaft is revolved on a horizontal plane, is used and the balancing machine operator indicates the amount of stock to grind off of the sides of the crank-arms from a previously calculated correction table. To facilitate balancing operations, Mr. Roberts states that closer machining limits were imposed and as a result the out-of-balance condition was greatly reduced. Crankshafts are balanced statically and dynamically which means they are in balance when at rest or when rotating. It is equally important to have connecting rods balance. The connecting rods for all aviation engines are selected in sets. The rods of a set ready to assemble into the engine should be alike within ± 1/8 ounce as to both total weight and center of mass. The rods are first weighed and classified for total weight and, as there is a difference of four ounces between the lightest and the heaviest rods, it is necessary to provide for 32 classes into which the rods are separated before proceeding with the next operation. This is to classify by the weight of the crankpin end only. The same limits, ± 1/8 ounce, are allowed as for the preceding operation. Special brackets were made and incorporated as units of a weighing scale, to make a fixture suitable for weighing the crankpin end of the rod. The scale is mounted on a surface-plate to which a bracket is attached also to support, off the scale, the piston-pin end of the connecting rod. A bracket fastened to the scale platform supports the crankpin end of the rod. Both brackets are equipped

with knife-edges, from which are suspended on rings certain U-shaped members. The upper arms of the U have knife-edges bearing on the rings, and the lower arms are made to fit loosely into the crankpin and the piston-pin holes of the connecting rod.

Firing Balance Important.—Firing balance, that is, having the intensity of the impulses as nearly equal as possible, contributes greatly to smoothness of operation. The combustion-chambers in the detachable heads of both the six-cylinder and the eight-cylinder engines are completely machined like their neighbors in the head, assuring uniform volume. Two cuts are necessary to obtain the fine finish required, it having been proved conclusively that a smooth surface greatly deters the formation of carbon deposits. A special machine is needed for profiling combustion-chambers. The machines used, of special design, were developed at the Packard plant. They have characteristics in common with automatic multiple-spindle profiling machines, the horizontal and transverse feeds for the cutters being actuated by two cams that control the contour of the chambers. The cutters are of special form and are of the end-milling type. In engines with composite cylinder construction, as radial air-cooled, the heads are completely machined before assembly and all heads are alike. Even one-piece cylinders can be completely machined so combustion spaces will all be equal.

A number of things can happen in assembling the engine which will offset all the careful effort spent in balancing the parts. To eliminate this, the pistons and the piston-pins are selected carefully as to both weight and fit. The fitting of the rings and the pins to the pistons and of the pistons to the cylinders requires extreme care; errors made at this stage of the assembly, such as bending the connecting rod slightly or distorting the pistons, will result in a number of serious engine troubles, among which are a loss of compression with consequent varying of the explosion pressures, oil-pumping and scoring of the cylinder-walls and the piston. Any of these will cause excessive vibration with its resultant noises and rough running. Cases have been known where vibration due to marked unbalance, such as losing part or all of a propeller blade, is enough to tear an engine away from its fastenings in the fuselage and have it drop to the ground.

Normalized Steel.—Heat treatment has always been an important factor in utilizing alloy steels successfully and unless great care is taken in heat treating processes, the great natural increase in physical properties of the materials used may be entirely lost and the steel will be no better than cheaper commercial metals.

The change in practice from heat-treating to normalizing operations for certain vital automotive parts, such as crankshafts and connecting rods, which is being introduced by the Vanadium Corp. of America, entails a thorough knowledge of the behavior of all steels when subjected to such normalizing operations. At first thought it might be assumed that if the quenching and drawing operations were done away with, almost any steel that would produce the necessary physical properties and hardness would be satisfactory, and the natural tendency would be to go to the steel of lesser cost. In promoting this radical change in production practice, the Vanadium Corp. feels that carbon vanadium steel is the ideal normalizing

steel. What the producer of automotive parts expects and strives to obtain in his heat-treating operations is uniformity of product, and he must expect to obtain this same uniformity in his normalizing operations. It might be thought that this uniformity should appear in all steels if simply annealed or normalized, but this is far from being true. For crankshafts a certain hardness and certain physical properties must be obtained, and an alloy steel of some type is desirable. Because of possible variation in rates of cooling that may be encountered from one end of the year to the other, as a result of changing weather conditions, etc., any element that has a tendency toward "air hardening" should be avoided. The commercial alloy steels that would naturally be considered are straight nickel steels, chrome nickel steels, straight chrome steels, chrome-molybdenum steels, nickel-molybdenum and high manganese steels, aside from the chrome-vanadium and carbon-vanadium steels. There are certain requirements or properties that the consumer must demand when using a normalized steel on a production basis, the principal ones being:

1. Physical properties.
2. Hardness and uniformity of hardness.
3. Uniformity and ease of machining and drilling.

If physical properties were the only requirement, several types of steel would answer the purpose. In dealing with automotive crankshafts—to which this article refers particularly—a minimum elastic limit of 70,000 pounds per square inch should be obtained, after heat treating or normalizing (as the case may be). Satisfactory values for the ultimate strength, elongation and reduction of area will naturally follow in the type of steel that would be considered, whether heat treated, that is quenched and drawn, or normalized.

Tests performed on various types of steel in the normalized state and of such analysis as to produce the minimum elastic limit of 70,000 pounds per square inch are given below. These tests were made on steel bars which had been forged and normalized as the parts would be on a production basis. It is a common experience that results of tensile tests performed on a set of test specimens will show considerable variations. The results given in table were obtained on a given test specimen and represent an average of what may be expected. The table also shows that if physical properties alone were used as a basis for adoption any one of the types could be considered. In the case of the straight chrome steel, a higher carbon content and, possibly, a little higher chromium content would bring the elastic limit up to requirements.

COMPOSITION AND PHYSICAL PROPERTIES OF ALLOY STEELS

ANALYSIS									Red	
C	Mn	V	Ni	Cr	Mo	El	Ult	Elong.	of Area	Brinell
.49	.80	.18			.31	75,000	160,250	28.5	60.1	207
.43	.72			.59		57,150	108,550	19.0	45.3	207
.47	.68		1.18	.70		68,400	120,600	19.5	55.2	252
.52	.57			.94	.31	86,200	142,500	18.0	59.4	309
.45	.89		1.80			65,000	118,250	23.0	54.8	217

It is interesting to note, however, from the values obtained in these particular tests that the Brinell hardness is quite misleading as an index of the elastic limit. As has been pointed out many times in other books it does follow the tensile strength fairly closely. It is also very interesting to note that any steel containing chromium, either alone or in combination with another element, and normalized, has a considerably greater hardness than a similar steel free from chromium and which has an equal or greater elastic limit. Note also the greater hardness of the straight nickel steel as compared with the carbon vanadium, even with a considerably lesser elastic limit. This brings out very clearly the air-hardening tendency of nickel and chromium alloys—a property very much to be avoided in normalized steels for production purposes. One of the important advantages of vana-

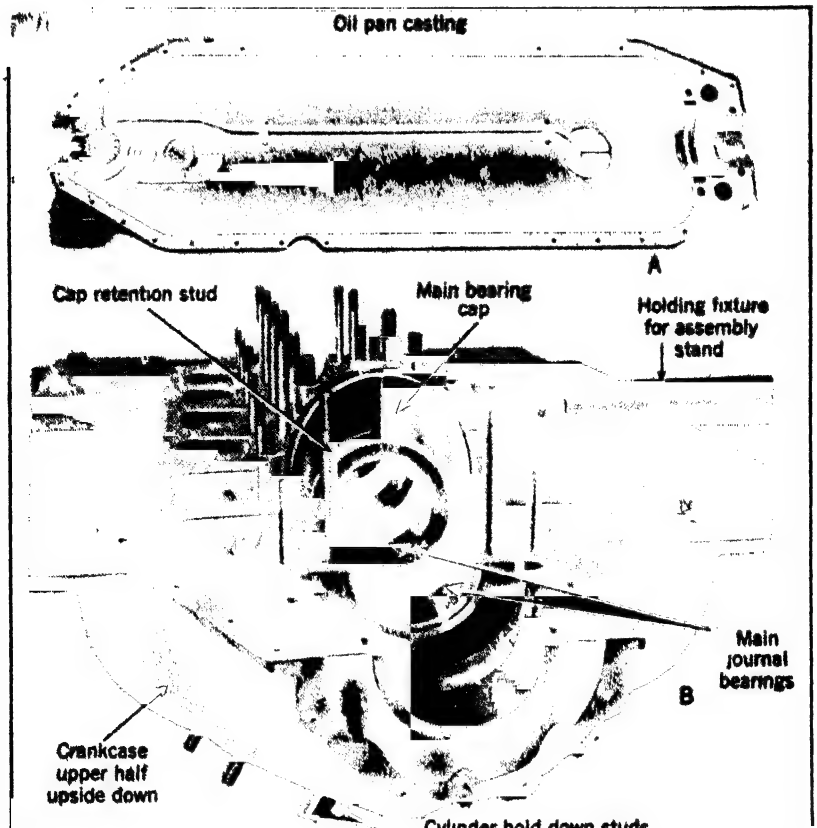


Fig. 358A.—View Showing Construction of the Crankcase of the Fiat A20 Aviation Engine. The Oil Pan Shown at A is Fastened to the Main Crankcase Member Shown at B.

dium in connection with the particular subject under discussion is that it does not confer marked air-hardening properties. Normalized plain vanadium steel has found considerable application in complicated shapes, such as motor crankshafts and many automotive forgings which must show a

high elastic limit without heat treating, since the latter causes distortion and a tendency to crack. Vanadium steels are characterized by exceptional fineness of grain. Either the trace of vanadium in the ferrite obstructs diffusion, or the vanadium-bearing cementite can diffuse less readily, so that the sorbite of vanadium steel tends to be finer, with less agglomeration of cementite and separation of ferrite than without the vanadium. This fine grain is undoubtedly the reason for the high elastic ratio of vanadium steel, being higher than that of most other common alloys.

Case-Hardened Crankshafts.—That case-hardened surfaces are much harder than oil-hardened surfaces and therefore much more resistant to wear is well known. The differences in the hardness numbers furnish a good index to the difference in the wearing qualities. Experience data relating to the superior wearing qualities of case-hardened as compared with the oil-hardened crankshafts have now been given out by the H. H. Franklin Mfg. Co., which has used case-hardened crankshafts since 1921. Since 1920, 1,434 crankshafts of the oil-hardened type were sent to the factory for regrinding, while since 1921 only thirteen case-hardened shafts were received for the same purpose, and all of these during the last three years, there having been not a single case of regrinding during the first four years that the case-hardened crankshafts were in use.

Camshaft Influence on Crankcase.—Before going into the subject of crankcase construction it will be well to consider camshaft location which is properly a part of the valve system and which has been considered in connection with the other elements which have to do directly with cylinder construction to some extent. Camshafts are usually simple members, and when carried at the base of the cylinder, in the crankcase of Vee-type motors, it is supported by suitable bearings. As previously mentioned this calls for tappet rod and rocker arm operation of the overhead valves. A typical camshaft design for engine base mounting is shown at Fig. 358. Two main methods of camshaft construction are followed—that in which the cams are separate members, keyed and pinned to the shaft, and the other where the cams are formed integral, the latter being the most suitable for airplane engine requirements. The camshaft shown is of the latter type, as the cams are machined integrally. In this case not only the cams but also the gears used in driving the auxiliary accessory drive shafts are forged integral. This is a more expensive construction, because of the high initial cost of forging dies as well as the greater expense of machining. It has the advantage over the other form in which the cams are keyed in place in that it is stronger, and as the cams are a part of the shaft they can never become loose, as might be possible where they are separately formed and assembled on a simple shaft. When the camshaft is carried in the engine case as in the Curtiss OX motor, provision must be made for its support. In most modern engines, the crankcase can be made very simple because the camshafts are driven by gears and mounted above the cylinder heads. A typical Vee engine crankcase with provisions for camshaft support is shown at Fig. 359.

Engine-Base Construction.—One of the important parts of the powerplant is the substantial casing or bed member, which is employed to support the cylinders and crankshaft and which is attached directly to the fuselage

by engine supporting members. This will vary widely in form, but as a general thing it is an approximately cylindrical member which may be divided either vertically or horizontally in two or more parts. Airplane engine crankcases are usually made of aluminum alloys, a material which has about the same strength as cast iron, but which only weighs a third as much. In many cases cast iron is employed in automobile motors, but is

Fig. 359.—Views of Upper Half of Crankcase Employed on Early Aviation Engine of the Eight-Cylinder Form.

not favored by airplane engineers because of its brittle nature, great weight and low resistance to tensile stresses. Where exceptional strength is needed alloys of aluminum bronze may be used, and in some cases where engines are produced in large quantities a portion of the crankcase may be a sheet steel or aluminum stamping.

Crankcases must always be large enough to permit the crankshaft and parts attached to it to turn inside with ample clearance and obviously its length is determined by the number of cylinders and their disposition. The crankcase of the single row, radial cylinder engine or double-opposed cylinder engine would be substantially the same in length, though the latter would be slightly wider to permit of off-setting the cylinder bores.

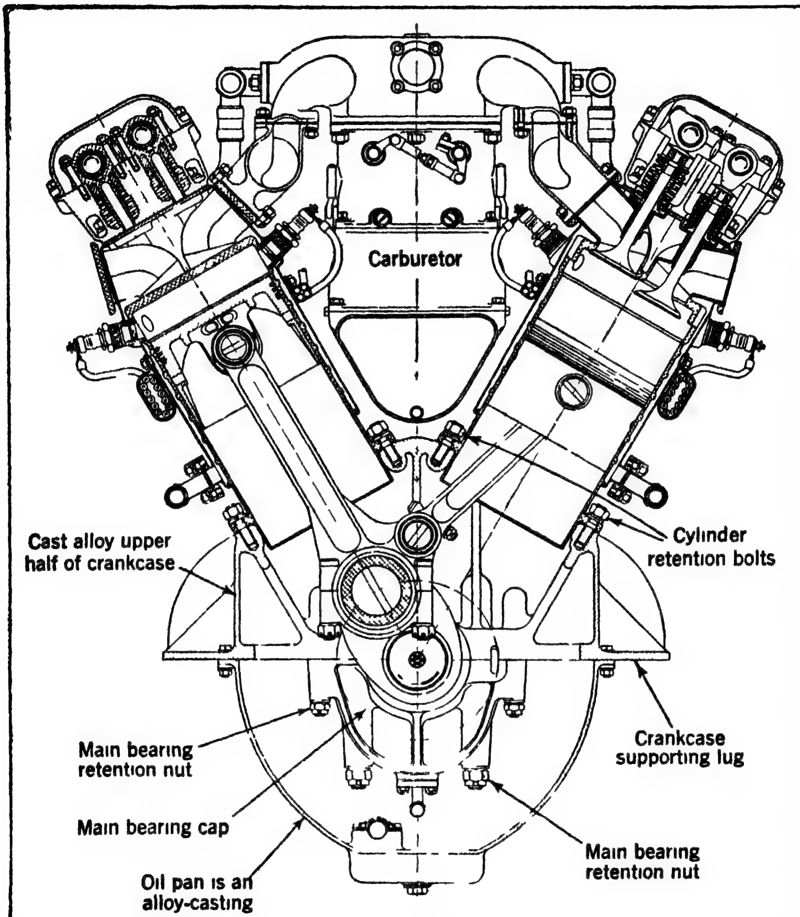


Fig. 359A.—Vertical Sectional Elevation of the Fiat A20 Aviation Engine, Showing the Method of Crankcase Construction. Note that the Cylinder and Crankshaft are Carried by Upper Half, While the Lower Portion Serves Merely as an Oil Pan. Attention is Directed to Substantial Construction of Main Bearing Caps.

That of a four-cylinder will vary in length with the method of casting the cylinder. When the four cylinders are cast in one unit, as in automobile practice, and a two bearing crankshaft is used, the crankcase is a very compact and short member. When a three-bearing crankshaft it utilized, the engine base is longer than it would be to support a block casting, but it is shorter than one designed to sustain individual air-cooled cyl-

inder castings and a five-bearing crankshaft. It is now common construction to cast an oil container integral with the bottom of the engine base and to draw the lubricating oil from it by means of a pump, as shown at Fig. 358 A. The arms by which the motor is supported in the fuselage are substantial-ribbed members cast integrally with the upper half as shown at Fig. 359 A which shows FIAT A 20 engine construction.

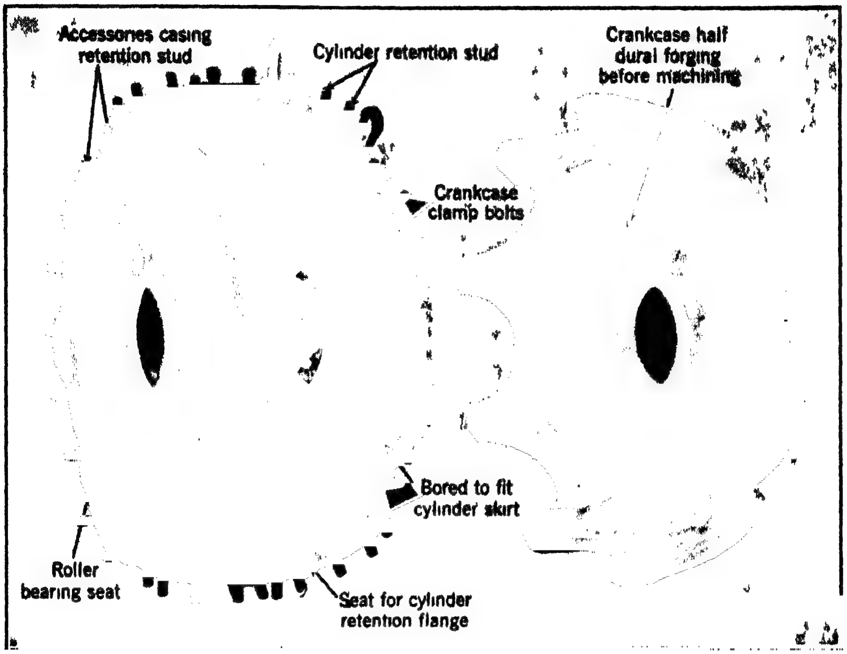


Fig. 360.—Crankcase of Pratt & Whitney "Wasp" Engine is Shown at the Left of the Illustration and is Made of Two Forged Duralumin Halves, One of which is Shown at the Right of the Illustration.

The approved method of crankcase construction favored by the majority of engineers is shown at Fig 358 B, bottom side up. The upper half not only forms a bed for the cylinder but is used to hold the crankshaft as well. In the illustration, the main bearing boxes form part of the case, while the lower bearings are in the form of separately cast caps retained by suitable bolts. In the construction outlined the bottom part of the case serves merely as an oil container and a protection for the interior mechanism of the motor. The cylinders are held down by means of studs screwed into the crankcase top, as shown at Fig. 359.

The simplicity of the crankcase needed for a revolving or static radial cylinder motor and its small weight can be well understood by examination of the illustration at Fig. 360, which shows the engine crankcase for the nine-cylinder Pratt & Whitney Wasp engine. This consists of two accurately machined duralminum forgings held together by bolts as clearly indicated. The crankcase of the Bristol Jupiter is also made of two main

portions, the front and rear half crankcase, which are of stout, well-ribbed section and machined from duralumin forgings, a face joint being made on the center line of cylinders by nine collar bolts, the rearward projections of which are used for attaching the engine to the airplane mounting. The crankcase of the early Gnome engine, which was machined from alloy steel forgings is shown at Fig. 362 B, and was the ancestral type of the later construction where "dural" forgings are employed or aluminum alloy castings in static radial engines.

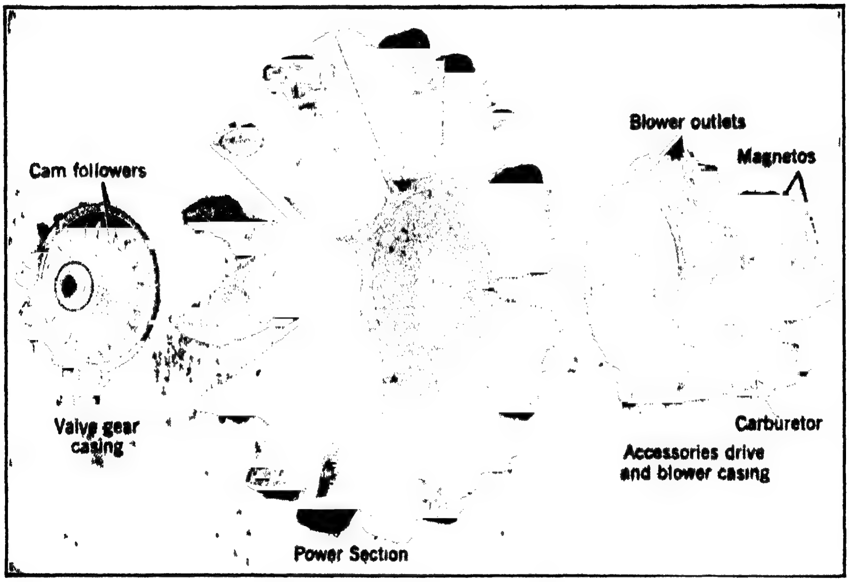


Fig. 361.—Crankcase Assembly of Pratt & Whitney "Wasp" Motor Has the Camcase Carrying the Valve Lifter Members at the Propeller End and Another Group in which All of the Accessory Drives and Superchargers are Installed, Bolted to the Anti-Propeller End.

Special Requirements Dictate Crankcase Design.—Crankcase construction has developed to a point where it can be said that each branch of the automotive industry has some special requirements which impose certain restrictions on the design of the crankcase. In early automobile engines the crankcase was made in two halves, and the crankshaft main bearings were contained half in the upper crankcase and half in the lower portion. This construction was very rigid and strong but made it impossible to adjust or replace these bearings without dismantling the whole engine. It then became the practice to mount the upper halves of the main bearings in the main part of the crankcase to which the cylinders were bolted, and the lower halves of the bearings were supported in bearing caps as shown at Figs. 358 and 359. The lower half of the crankcase then became an oil-pan and could be made fairly light. For purposes of inspection the oil-pan could be removed without taking the engine out of the car. This represented

a distinct advance from the standpoint of maintenance and is standard practice today in automobile manufacture. When aviation engines are overhauled, however, they are invariably removed from the fuselage, except in the case of a top overhaul, in which the bearings are not repaired, so the stronger construction is favored by designers. Some of the newer water-cooled aviation engines have the main bearings suspended from the top half of the case and use the oil-pan system common in motor cars.

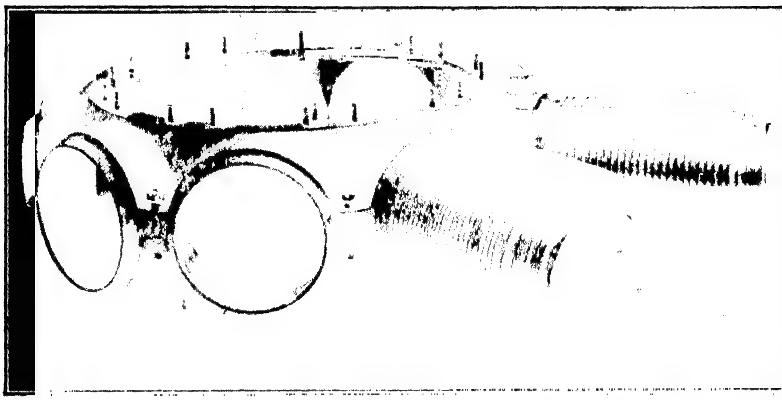


Fig. 362B.—Simple and Compact Crankcase Utilized on "Gnome" Engine is the Ancestral Type of Many Static Radial Engine Designs.

Aircraft engine crankcases when cylinders are in line are nearly always built according to the original automobile practice; that is, the crankcase is divided in a plane through the center of the crankshaft and the main bearings are carried between the two halves of the crankcase. This construction is favored since it is the lightest and most rigid construction, although in the matter of accessibility there would be still much to be desired if the common practice was to leave the engine in the fuselage when overhauling it, which is not the case. This is shown at Fig. 363.

In marine engines both forms of crankcase construction already referred to are used but, in addition, large hand-hole covers are provided on the sides of the upper half of the crankcase to allow connecting rod bearing inspection and even piston removal in some cases. This matter of accessibility in a marine engine is very important, since the engine cannot be readily removed and repairs must sometimes be made far from any base. This type of crankcase construction is sometimes used in large aviation engines intended for use in dirigible balloons.

Radial Engine Crankcase.—The construction of the crankcases of radial-cylinder engines is greatly simplified by having the valve actuating mechanism carried in another casing bolted to the main casing and grouping all the accessories and their drive on still another case or assembly. The method of construction shown at Fig. 361 is followed by the Pratt & Whitney Aircraft Corporation and as a result, it manufactures two engines, one of greater horsepower than the other, 80 per cent of the parts of both engines being interchangeable, a manufacturing advantage of great value. Not all radial engine crankcases are made in two halves bolted together

on the engine center line. The Wright Whirlwind crankcase is a one-piece aluminum casting having extensions for valve operating mechanism and fuel mixture supply and accessory drives cast integrally. All such cases have bolted-on cover plates.

Packard X Crankcase.—The Packard X engine, which is really a radial engine even though it has four banks of six cylinders each, uses a simple barrel type crankcase shown at Fig. 362 A and has the accessory drive and timing gears housed in separate cases, bolted to the main crankcase which is of barrel form. The construction of the main bearings is unusual and at first glance, excellent designers might consider them a poor design because of their large diameter or bore and consequently high rubbing speeds.

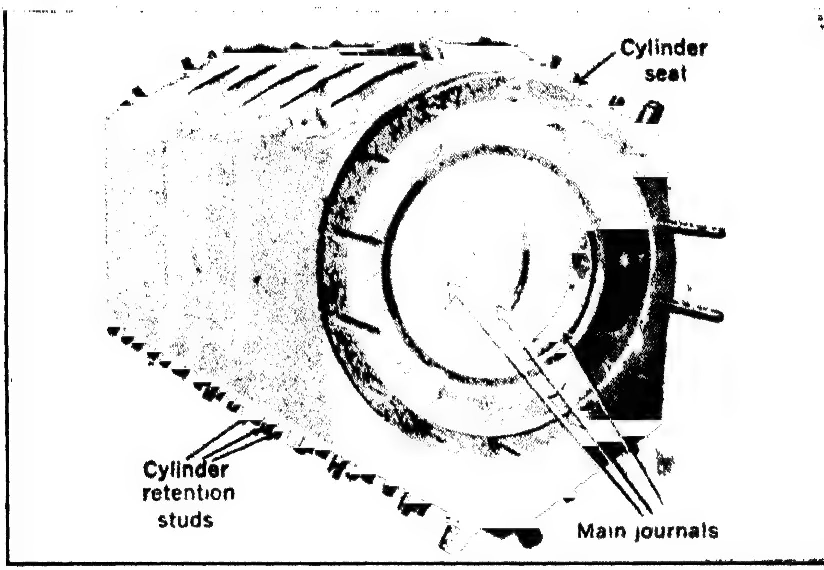


Fig. 362A.—Barrel Type Crankcase Used on the 24-Cylinder Packard X Engine.

The extreme ruggedness and simplicity of design are apparent. In general, the Packard X crankcase consists of a hexagon-shaped aluminum alloy casting with seven large main-bearing shell bores formed in well-ribbed diaphragms. The main bearings are babbitt-lined steel shells of 7¾-inch inside diameter. These bearings are shrunk into the crankcase, which is previously heated to 212 degrees Fahrenheit. The bearings are supported on a water-cooled mandrel and slide easily into place, being then retained by cap screws. After the water-cooled mandrel is withdrawn and the crankcase cools to operating temperature, these bearings naturally are firmly held in place. Although it is believed that the surface speed to which these bearings are subjected far exceeds previous practice in this regard, no trouble has been experienced with them. The surface speed is about 5,500 feet per minute, and the maximum load on the bearings is somewhat less than 700 pounds per square inch. The necessity for this type of crankshaft in which the crank cheeks perform the function of main journals, or

vice-versa, is brought about by the need for economizing in the over-all length of the engine. Each bank of cylinders has the same dimension between adjacent cylinder centers as that used in the Model 1,500 twelve-cylinder engine. Doubling the power to be transmitted by the crankshaft, with the consequent increase in bearing loads and crankshaft stresses, could be met only by the very unconventional means employed. It is interesting to note that this crankshaft weighs 161 pounds, or about double that of the Model 1,500 crankshaft.

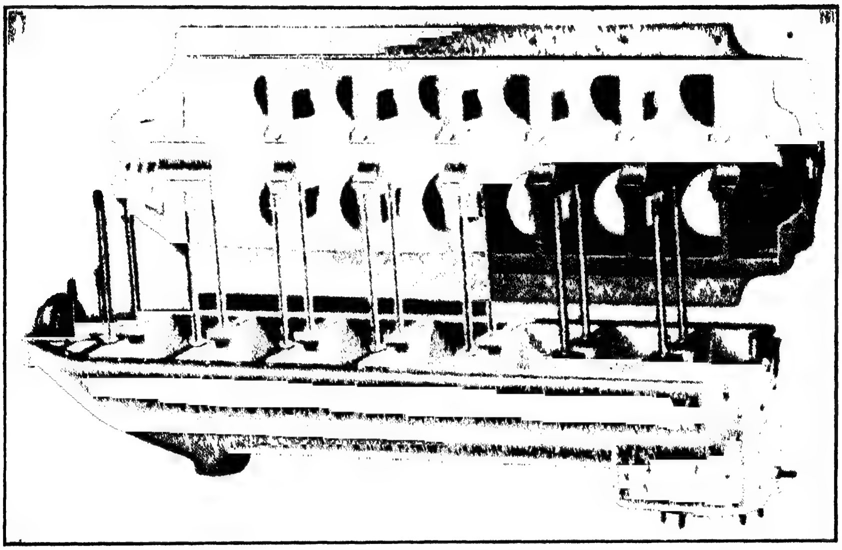


Fig. 363.—Illustration Showing Method of Construction of the Liberty Engine Crankcase. Note Method of Fastening Upper and Lower Halves Together by the Use of Long Through Bolts Extended to the Top of the Upper Crankcase, as Well as by the Numerous Smaller Bolts Holding Flanges Together.

Anti-Friction Bearings.—Roller-bearings may be made with either solid or flexible rolls. A solid roller should be short in order to prevent deflection or distortion of the roll if the stress is not evenly applied. If long solid rollers are used, one end will tend to travel faster than the other, and this is apt to produce friction between the rolls and the retaining cage. It is contended by adherents of the flexible roll construction that this type adapts itself more readily to irregularities in shaft contour, and thus turns with less friction than do the solid rolls. The flexible roller-bearings are made in two types, one employing long steel rolls, while the high duty type uses shorter rolls of high-tensile strength alloy steel and accurately ground inner and outer race members. In the regular or long roll pattern the shaft itself is intended to form one bearing surface, and as this is generally soft, the rolls must be long in order to distribute the load over more surface. Flexible roller bearings are wound from strip stock, so that right and left spirals exist in alternate rollers, and the oil will be constantly circulated from one side of the bearing to the other. They are seldom used in aviation engines, rollers of solid section and short in length but of considerable

diameter being favored. The carrying capacity of a short roll of the flexible type is less than a solid roller of the same length and diameter.

In either ball- or roller-bearings the effect of the load is to flatten the supporting members a very small amount, so that in a roller-bearing the contact between the roller and race-way may be represented by a rectangle having a length equal to the roll, but with a width so small that it is usually considered a line contact. In a ball-bearing the flattening of the ball produces an ellipse of such small area that it is usually considered a point of contact. As the reduction of friction depends upon the amount of surface in contact, it can readily be understood that the form having the least amount of surface would have least friction. Ball-bearings are generally employed where it is desirable to reduce friction to a minimum or where bearings must attain high speed. Roller-bearings have been widely applied where bearings of relatively small diameter but of large carrying capacity must be used, where the shafts revolve at a comparatively low speed, and where maximum bearing efficiency is not important. Many examples showing the practical application of all types of anti-friction bearing will be found at various points in this work.

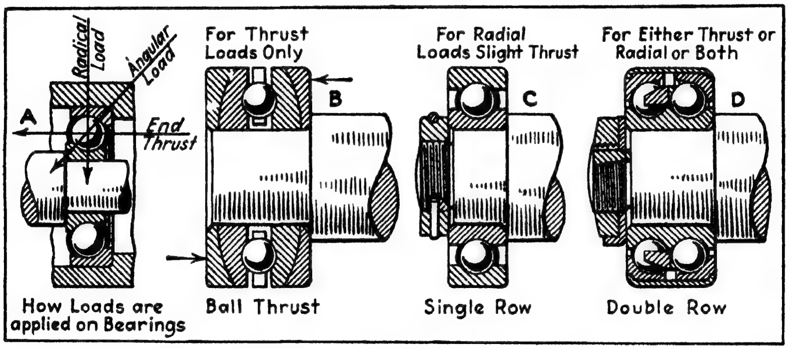


Fig. 364.—Types of Ball Bearings Used in Motor Construction.

An angular resultant is a load composed of a combination of end thrust and radial stress. A ball- or roller-bearing having angular line of contact is suited to resist either a radial or thrust load or a combination of the two. The angularity of a load line of a bearing affects radial and end thrust capacity, depending upon the angle the load line assumes with the horizontal or vertical center lines of the bearing. As a bearing with a vertical load line has the greatest radial capacity and practically no resistance to end thrust, it follows that one with a horizontal load line will have the greatest thrust capacity and practically no resistance to radial loads. This is clearly shown at Fig. 364 A. While it may be thought at first glance that a 45 degree angle load line would give equal thrust and radial capacity, this is not true as only the balls or rollers in half the bearing carry radial load while all of them resist end thrust.

Great care is needed to keep anti-friction bearings free from water or grit, and also to keep them adequately lubricated. The lubricant must be a pure mineral substance, and should not contain acid or animal fillers,

because the highly polished surfaces of the anti-friction members and races will be roughened by etching or rusting due to chemical action, and this will interfere with smooth operation and tend to produce rapid depreciation. A heavy bodied lubricant of the semi-fluid type is best adapted for use in bearings subjected to heavy loads and revolving at low speed as in gear housings but as the most common application in aviation engines calls for their use in the engine interior where the lubrication and dirt protection features are properly taken care of and only oil of the best quality used, they work perfectly without attention.

If an anti-friction bearing is housed in an oil-retaining, dust-excluding housing, it is good practice to use a medium grade oil and operate the bearings in a constant bath of lubricant. Ball- or roller-bearings should never be driven in place or removed with a steel drift or hammer, because the races are hard steel and are apt to be cracked unless they are forced in place either with a steady pressure, as by an arbor press, or by blows applied through the medium of a block of hard wood or piece of babbitt metal interposed between the hammer and the bearing. The blow should always be directed or the pressure exerted against the race member that is being forced in place and never transmitted through the balls or rolls from one race to the other.

Lightness of Construction.—The development of the aeronautic engine has focused the attention of designers and the public upon the light engine. In the case of the aeronautic engine, lightness is obtained mainly through machining out low-stressed portions of the various members concerned and because the large size of these engines renders them less susceptible to the limitations imposed upon lightness by foundry and forge considerations. It is not necessary in aviation engines to use twice as much material as is required to make a part which would work perfectly, providing it could be got in place without its being broken by dropping upon the shop floor. In the case of automotive engines generally, such methods of obtaining lightness are either precluded or prohibitive in cost, though widely used in the design and production of aviation engines. Apart, however, from the use of material of high specific strength and low weight which side-tracks foundry and forge difficulties, there remains the consideration in the light of recent knowledge of the all-important question of engine proportions from the point of view of least crankcase weight. It is obvious that if the mean effective pressure is independent of the stroke-bore ratio, the same power will be developed by engines of the same cylinder capacity at the same speed. Aeronautic engine experience throws considerable light on this question, some of the latest engines being of the short-stroke type and developing mean effective pressures as high as those with longer strokes. In aeronautic engines overhead valves are universally used, so that adequate turbulence is obtained in the combustion-chamber, due to the compact shape of the latter. In the case of automobile engines it is undoubtedly easier to obtain turbulence in long-stroke engines with side valves, than in short-stroke engines with side valves, but the difference is only a small percentage and there is, as already indicated, reason to believe that the combustion-chamber of a side-valve engine by using the Ricardo head design can be modified so as to negative entirely its apparent

deficiency in respect to turbulence.

If, then, the problem of stroke-bore ratio can be denuded of its power aspect, which also carries with it thermal efficiency, the ratio of stroke to bore can be settled on the basis of minimum crankcase weight and manufacturing convenience. From these points of view some authorities believe the short-stroke engine has everything in its favor. To begin with, the overall length of the engine is usually settled by the summation of the valve diameters, which are necessarily settled by the cylinder capacity, being the same for both long- and short-stroke engines of the same capacity. The overall crankshaft length, therefore, is also settled, since the bearing lengths should be proportional to the cylinder capacity, which is independent of the stroke-bore ratio. On the other hand, the larger throw of the crank of the long-stroke engine increases the weight directly, while its inherent extra "crankiness" calls for larger dimensions, if equal stiffness and freedom from vibration are to be assured. Following the extra crank-throw is the extended section of the crankcase, necessitating extra ribbing and metal for strength and stiffness, extra height on the cylinders due to the longer stroke and longer connecting rods, if the ratio of crank length to connecting rod length is to be the same as in the short-stroke engine. The length of valves and camshaft center distance from crankshaft are also increased in the long-stroke engine, the latter calling for considerably heavier timing-gear than otherwise required.

Materials Used in Engines.—The constructors of aviation engines have evidently determined that some materials are more suitable than others, and in most instances there is a similarity in details of construction which founds the belief standard methods will be adopted and accepted generally, as has been the experience in motor car building. The suitability of the materials ordinarily used may be determined by any of several methods of computing stress, which may be divided into longitudinal and transverse. Under the former head may be considered tensile strength or the power of resisting pulling force, and compression strength or resistance to crushing force. Under transverse stress may be considered shearing, or resistance to cutting across; bending, which is resistance to cross breaking, and torsional strain, which is obtained by resisting twisting stress. When a load is applied to any structure or component it causes a change of form, which may be very slight, but which always takes place, however small the load. The change of form thus produced is called strain and the acting force stress.

The ultimate strength is the maximum resistance under load, the greatest stress that can be obtained or exist before rupture. The working load is that which the piece is proportioned to bear. In proportioning any structure, engineers usually allow a certain factor of safety, in machinery this being six or eight, while in structures erected by the architect or civil engineer, the figure is five or six. The factor of safety should be larger under live and intermittent loads than under steady or dead loads. This means that if a certain piece of metal appears to be theoretically strong enough of a certain section to do stated work in practical application it should be made five or six times stronger than actually necessary to insure an ample margin of safety for unexpected stresses. The material which best resists

the strains imposed upon it will be most suitable, and for the airplane engine that which has the greatest strength per unit weight is the most satisfactory, providing that it has sufficient capacity. Airplane safety factors average around eight.

When using metal, lightness may often be obtained by the use of pressed forms and stampings in place of the heavier castings and forgings, which do not possess greater strength or rigidity in many instances. Tubing may be used in structural work, this made of light and highly resistant materials and strength is obtained with a minimum amount of metal. When a forging or stamping of metal is used it may be lightened by boring holes without too great a sacrifice of strength, such holes of course being bored only in the lightly loaded portions of the piece. It will be apparent that alloy steels, which have excellent physical properties and great strength per unit weight or area of section are the best for use in aerial motors. Less material is needed in a piece to secure adequate strength and weight is correspondingly reduced. The range of alloy steels is very great and numerous alloying elements are used in connection with ferrous metals. These include nickel, chromium, vanadium, manganese, molybdenum, tungsten, silicon and various combinations of these metals. Most of the alloy steels are costly to buy, and expensive to machine and heat treat.

More than 40 different kinds of material are used in the modern aircraft engine. The most interesting materials are the light and unusual alloys. Magnesium probably is one of the more interesting of these to engineers who are not connected with the aircraft industry. This alloy is 40 per cent lighter than aluminum, has an average tensile strength of 20,000 pounds per square inch and an elongation of four per cent. It is comparable in strength and elongation with the best non-heat-treated commercial aluminum crankcase alloys. The lighter alloys of magnesium, alloyed with about five per cent of aluminum, are much less subject to corrosion than the original alloys that were produced. This alloy has been cast in almost every form that is used in an aircraft engine; however, the present cost of production prohibits its use in quantity. It is hoped that in the future sufficient quantities will be available so that the material can be used with success in later models of production engines. With this material available economically and with higher engine speed, remarkable performances may be expected.

Aluminum-bronze is another material that has come into extensive use in aircraft engines. This makes a very remarkable valve insert metal, as its coefficient of expansion is about the same as that of aluminum. The valve seat, after a few hours of engine running, presents a mottled appearance almost as if the metal were pitting. We find, however, that this is not the case and to recut the seats because of this appearance is not wise. The valves will continue to function for hundreds of hours without trouble with this construction if the proper cooling is provided around the valve insert. The material can be heat-treated to give about 200 Brinell hardness for this work.

Duralumin is an alloy with which nearly everybody in the automotive industry today is familiar. It is used extensively in modern engines, in both

Properties of Aluminum Alloys Used for Aircraft Engine Construction

Name of Alloy	Approximate Chemical Composition %	Heat Treatment	Average Physical Properties: Tensile Strength lb. sq. in.	Elongation %	Brinell Hardness No.	Remarks
8% Copper Alloy W. A. C. NF-104	Copper 8.0 Impurities 1.7 max. Aluminum remainder	Anneal	14,000 to 18,000	5	55 to 60	Used for general casting purposes.
Silicon Alloy (Developed by U. S. Air Service)	Copper 3.0 Silicon 4.0 Impurities 2.0 max. Aluminum remainder	Anneal	20,000 to 23,000	2.0 to 3.0	55 to 65	Easy to cast and free from casting defects. Does not machine freely.
Lynite No. 195 W. A. C. NF-130	Copper 5.0 Manganese .1 Silicon .65 Impurities 1.0 max. Aluminum remainder	36 hrs. at 975° F. Quench cold water	25,000 to 35,000	8.0	50 to 65	Used for highly stressed parts not subject to heat.
Duralumin (Forged)	Copper 4.0 Manganese .7 Magnesium 1.5 Impurities 1.0 max. Aluminum remainder	Varies with manufacturer	55,000 to 60,000	17.0	100	For parts subject to high stresses and shock loading which can be made by forging or rolling.
"Magnalite" or "Y" Alloy W. A. C. NF-125	Copper 5.5 Magnesium 1.5 Nickel 2.25 Impurities 2.0 max. Aluminum remainder	3 hrs. at 950° F. Quench boiling water. Age 16 hrs. at 300° F.	25,000 to 40,000	.5 to 3.0	90 to 110	A high strength cast and heat treated alloy. Used for cylinders, pistons, etc.
Al-Cu-Fe-Mg Lunite No. 122	Copper 10.0 Iron 1.25 Magnesium .25 Impurities 1.0 max. Aluminum remainder	5 hrs. at 925° F. Quench boiling water. Age 16 hrs. at 300° F.	30,000 to 40,000	.... 1.0	90 to 110	Used principally for pistons.
Magnesium Alloy	Magnesium 96% Aluminum 3% Zinc 1%	None	19,000	6.0	45	Experimental.

the cast and forged forms. It is particularly advantageous in the forged form owing to the small amount of work that is necessary on the forgings after they are received. The finish of the forgings is perfectly satisfactory and it is necessary only to remove the flash before putting the forgings into the machine shop. We have found that the cast high-tensile aluminum alloy is unusually satisfactory for cylinder heads. The average tensile strength is more than 30,000 pounds per square inch and elongation about six per cent. It is extremely nonporous and very uniform in texture, and it machines very satisfactorily. The cylinder heads and water jackets of our water-cooled engines and in the crankcases of the air-cooled engines are the places where this alloy is used. It is approximately ten per cent lighter and nearly 100 per cent stronger than the usual crankcase aluminum alloys.

Y-alloy is an alloy of aluminum with copper, nickel and magnesium. It is used for pistons in many engines and in the cylinder heads of air-cooled engine. This material is a light aluminum alloy owing to its low copper content and is a wonderful bearing material. It seems to be as strong as high-tensile strength aluminum alloys in heat-treated condition but has a marked advantage in having strength at high temperatures. Y-alloy needs heat-treating operations to realize its properties to the utmost. High silicon aluminum alloys are used for sections varying from $\frac{3}{32}$ inch to $\frac{1}{8}$ inch in thickness successfully.

SCREW STOCK

S. A. E. Steel No.	Carbon Range	Manganese Range	Phosphorus, Max.	Sulphur Range
1112	0.08-0.16	0.60 0.80	0.09-0.13	0.075-0.15
1120	0.15-0.25	0.60 0.90	Max. 0.06	0.075-0.15

[1] The silicon content for steels No. 1350 and 1360 shall not exceed 0.30 per cent.

STEEL CASTINGS

S.A.E. Steel No.	Carbon	Phosphorus, Max.	Sulphur, Max.
1235	As required by physical properties	0.05	0.05

S. A. E. Steel Specifications.—Definite applications of S. A. E. Steels are not covered hereinafter, as it will be readily appreciated that the selection of a proper steel for a given part must depend upon an intimate knowledge of a number of important factors, such as the availability and price of the material, the detailed design of the part and the severity of the service to be imposed, whether the part is to be forged or machined, machineability and the method of manufacture. Only after a careful consideration of

these factors can the proper steels for the great variety of automotive parts be selected. By providing charts that show the conservative physical properties of the S.A.E. Steels, the designer is supplied with data that will in most cases enable him to prepare a list of satisfactory steels from the standpoint of physical properties, after which the final choice will depend upon such local conditions as machineability, heat-treating, delivery and price. The alloys of iron are numerous and varied and the best results can only be obtained by using heat treatments best adapted for the work the part is to do. Some pieces of the engine must be strong and tough, others must have a hard surface to resist wear. The following chemical compositions are given to show only a few of the possible alloys but those given have been definitely applied to automotive parts.

CHEMICAL COMPOSITIONS

S. A. E. Standard

CARBON STEELS

S. A. E. Steel No.	Carbon Range	Manganese Range	Phosphorus, Max.	Sulphur Max.
1010	0.05-0.15	0.30-0.60	0.045	0.05
1015	0.10-0.20	0.30-0.60	0.045	0.05
1020	0.15-0.25	0.30-0.60	0.045	0.05
1025	0.20-0.30	0.50-0.80	0.045	0.05
1030	0.25-0.35	0.50-0.80	0.045	0.05
1035	0.30-0.40	0.50-0.80	0.045	0.05
1040	0.35-0.45	0.50-0.80	0.045	0.05
1045	0.40-0.50	0.50-0.80	0.045	0.05
1046	0.40-0.50	0.30-0.50	0.045	0.05
1050	0.45-0.55	0.50-0.80	0.045	0.05
1095	0.90-1.05	0.25-0.50	0.040	0.05
1350[1]	0.45-0.55	0.90-1.20	0.040	0.05
1360[1]	0.55-0.70	0.90-1.20	0.040	0.05

NICKEL STEELS

S. A. E. Steel No.	Carbon Range	Manga-nese Range	Phos-phorus, Max.	Sulphur, Max.	Nickel Range
2315	0.10-0.20	0.30-0.60	0.04	0.045	3.25-3.75
2320	0.15-0.25	0.50-0.80	0.04	0.045	3.25-3.75
2330	0.25-0.35	0.50-0.80	0.04	0.045	3.25-3.75
2335	0.30-0.40	0.50-0.80	0.04	0.045	3.25-3.75
2340	0.35-0.45	0.50-0.80	0.04	0.045	3.25-3.75
2345	0.40-0.50	0.50-0.80	0.04	0.045	3.25-3.75
2350	0.45-0.55	0.50-0.80	0.04	0.045	3.25-3.75
2512	max. 0.17	0.30-0.60	0.04	0.045	4.50-5.25

NICKEL-CHROMIUM STEELS

S. A. E. Steel No.	Carbon Range	Manganese Range	Phosphorus, Max.	Sulphur, Max.	Nickel Range	Chromium Range
3115	0.10-0.20	0.30-0.60	0.04	0.045	1.00-1.50	0.45-0.75
3120	0.15-0.25	0.30-0.60	0.04	0.045	1.00-1.50	0.45-0.75
3125	0.20-0.30	0.50-0.80	0.04	0.045	1.00-1.50	0.45-0.75
3130	0.25-0.35	0.50-0.80	0.04	0.045	1.00-1.50	0.45-0.75
3135	0.30-0.40	0.50-0.80	0.04	0.045	1.00-1.50	0.45-0.75
3140	0.35-0.45	0.50-0.80	0.04	0.045	1.00-1.50	0.45-0.75
3215	0.10-0.20	0.30-0.60	0.04	0.040	1.50-2.00	0.90-1.25
3220	0.15-0.25	0.30-0.60	0.04	0.040	1.50-2.00	0.90-1.25
3230	0.25-0.35	0.30-0.60	0.04	0.040	1.50-2.00	0.90-1.25
3240	0.35-0.45	0.30-0.60	0.04	0.040	1.50-2.00	0.90-1.25
3245	0.40-0.50	0.30-0.60	0.04	0.040	1.50-2.00	0.90-1.25
3250	0.45-0.55	0.30-0.60	0.04	0.040	1.50-2.00	0.90-1.25
3312	max. 0.17	0.30-0.60	0.04	0.040	3.25-3.75	1.25-1.75
3325	0.20-0.30	0.30-0.60	0.04	0.040	3.25-3.75	1.25-1.75
3335	0.30-0.40	0.30-0.60	0.04	0.040	3.25-3.75	1.25-1.75
3340	0.35-0.45	0.30-0.60	0.04	0.040	3.25-3.75	1.25-1.75
3415	0.10-0.20	0.45-0.75	0.04	0.040	2.75-3.25	0.60-0.95
3435	0.30-0.40	0.45-0.75	0.04	0.040	2.75-3.25	0.60-0.95
3450	0.45-0.55	0.45-0.75	0.04	0.040	2.75-3.25	0.60-0.95

MOLYBDENUM STEELS

S. A. E Steel No.	Carbon Range	Manganese Range	Phosphorus, Max.	Sulphur, Max.	Chromium Range	Nickel Range	Molybdenum Range
4130	0.25-0.35	0.40-0.70	0.04	0.045	0.50-0.80	....	0.15-0.25
4140	0.35-0.45	0.40-0.70	0.04	0.045	0.80-1.10	....	0.15-0.25
4150	0.45-0.55	0.40-0.70	0.04	0.045	0.80-1.10	....	0.15-0.25
4615	0.10-0.20	0.30-0.50	0.04	0.045	....	1.25-1.75	0.20-0.30

CHROMIUM STEELS

S. A. E. Steel No.	Carbon Range	Manganese Range	Phosphorus, Max.	Sulphur, Max.	Chromium Range
5120	0.15-0.25	0.30-0.60	0.04	0.045	0.60-0.90
5140	0.35-0.45	0.50-0.80	0.04	0.045	0.80-1.10
5150	0.45-0.55	0.50-0.80	0.04	0.045	0.80-1.10
52100	0.95-1.10	0.20-0.50	0.03	0.030	1.20-1.50

CHROMIUM-VANADIUM STEELS

S. A. E. Steel No.	Carbon Range	Manganese Range	Phosphorus, Max.	Sulphur, Max.	Chromium Range	Vanadium	
						Min.	Desired
6120	0.15-0.25	0.50-0.80	0.04	0.04	0.80-1.10	0.15	0.18
6125	0.20-0.30	0.50-0.80	0.04	0.04	0.80-1.10	0.15	0.18
6130	0.25-0.35	0.50-0.80	0.04	0.04	0.80-1.10	0.15	0.18
6135	0.30-0.40	0.50-0.80	0.04	0.04	0.80-1.10	0.15	0.18
6140	0.35-0.45	0.50-0.80	0.04	0.04	0.80-1.10	0.15	0.18
6145	0.40-0.50	0.50-0.80	0.04	0.04	0.80-1.10	0.15	0.18
6150	0.45-0.55	0.50-0.80	0.04	0.04	0.80-1.10	0.15	0.18
6195	0.90-1.05	0.20-0.45	0.03	0.03	0.80-1.10	0.15	0.18

TUNGSTEN STEELS

S. A. E. Steel No.	Carbon Range	Manganese, Max.	Phosphorus, Max.	Sulphur, Max.	Chromium Range	Tungsten Range
71360	0.50-0.70	0.30	0.035	0.035	3.00 4.00	12.00-15.00
71660	0.50-0.70	0.30	0.035	0.035	3.00-4.00	15.00-18.00
7260	0.50-0.70	0.30	0.035	0.035	0.50-1.00	1.50- 2.00

SILICO-MANGANESE STEELS

S. A. E. Steel No.	Carbon Range	Manganese Range	Phosphorus, Max.	Sulphur, Max.	Silicon Range
9250	0.45-0.55	0.60-0.90	0.045	0.045	1.80-2.20
9260	0.55-0.65	0.60-0.90	0.045	0.045	1.80-2.20

Nickel and Chrome Nickel Steel Airplane Engine Parts.—Alloy steels containing chromium and nickel have been widely used in airplane engine parts. The steels used in the construction of the Wright "Whirlwind" engine parts where exceptional strength combined with light weight is desired are of the nickel or chromium alloyed types. The nine cylinders are bolted to the aluminum alloy crankcase with chrome-nickel steel studs and nuts. The single-throw crankshaft is made of S. A. E.-3,140 chrome-nickel steel and is mounted on two roller bearings and one outboard ball bearing which also takes the propeller thrust. The shaft is counter-balanced, the weights being securely fastened by means of rivets and bolts made of 3,140 chrome-nickel steel. The master rod and cap and bolts are made of chrome-nickel steel. The rod is fitted with a steel back babbitted bearing lubricated by pressure feed. The eight articulated rods are of 3,140

chrome-nickel steel. The piston pins are made of either 3,250 chrome-nickel or 6,150 chrome-vanadium steel, oil hardened, while the knuckle pins are made of 3,115 chrome-nickel steel, case hardened.

The crankshaft gear is made of 3,115 chrome-nickel steel, case hardened, and is keyed to the shaft. This part is made with a long sleeve which forms a bearing for the cam hub which is run concentric with the shaft. The inlet and exhaust cams are integral and made of 3,115 steel. The cams operate the roller tappets of 3,140 which are arranged radially opposite each cylinder. The push rods are of thin-walled 3½ per cent nickel steel tubing while the valve rockers are made of 2,330 steel. The magneto drive gears and coupling, the fuel pump gears, the tachometer gears and shafts are all chrome-nickel steel 3,140.

A complete list of the parts in which nickel or chrome-nickel steels are used is as follows:

Name of Part	S.A.E. Number
Balance weight pins	3140
Counterweight rivet	3140
Knuckle pin	3115
Knuckle pin lock	3140
Piston pin	3250 or 6150
Oil pressure relief body	3140
Oil pump drive gear	3140
Oil pump idler gear	3140
Oil pump gears	3140
Oil and fuel pump drive gear	3115
Tachometer drive gear	3140
Tachometer drive shaft	3140
Master rod	3140
Master rod cap	3140
Master rod bolts	3140
Articulated rods	3140
Crankshaft	3140
Crankshaft gear	3115
Magneto coupling drive gear	3140
Magneto coupling gear	3240
Magneto drive gear	3140
Magneto gear	3140
Magneto cap screws	3140
Cam driving pinion	3115
Cam drive gear	3140 or 6150
Cam driving gear	3240
Cam driving pinion thrust washer	3140
Inlet and exhaust cam	3115
Valve tappet roller pin	3115
Valve tappet	3140
Valve tappet ball socket	3140
Valve push rod	3½% Ni seamless tubing
Valve push rod adjusting nut	3140 or 3250
Valve rockers (inlet and exhaust)	2330
Valve rocker support pin sleeve	3140 or 3250
Valve rocker support pin	3140
Valve rocker support stud nut	3140
Valve spring washer (lower)	3140
Valve rocker support cover screw	3140
Valve rocker support pin spacer	3250 or 6150
All important screws, studs, bolts and nuts.	

The specifying of proper materials and heat treatments on a motor of this type is of the greatest importance and is vital to its reliability and long-life. Chrome-nickel and nickel steels have been selected largely on account of their uniform and reliable properties. The forging grade 3,140 used in so many parts is heat treated to give a combination of high strength and toughness. The 3,115 steel used in carburized parts gives a high hardness of the case combined with a strong and tough core.

Although the S. A. E. numbers have been used to indicate the steel, the specifications of the Wright Company often vary from them slightly to insure the highest grade material. The chemical analyses required on 3,115 and 3,140 are given below:

	3115	3140
Carbon	0.10 — 0.20	0.35 — 0.45
Manganese	0.50 — 0.80	0.50 — 0.80
Phosphorus	Max. 0.04	Max. 0.04
Sulphur	Max. 0.045	Max. 0.045
Nickel	1.10 — 1.50	1.10 — 1.50
Chromium	0.45 — 0.75	0.45 — 0.75

Some typical heat treatments used in various parts are given below:

Heat treatment and operations of inlet and exhaust cam, made of 3,115 steel, case hardened:

1. After forging, normalize at 1,625-1,650 degrees Fahrenheit.
2. Machine.
3. Copper plate except gear teeth and cam.
4. Carburize at 1,650-1,675 degrees Fahrenheit to secure a depth of case of .050 to .060 inch; cool in the pot.
5. Harden and examine test pieces of the same steel to verify depth of case.
6. Drill holes.
7. Heat to 1,400-1,425 degrees Fahrenheit; quench in water in a fixture to prevent warpage.
8. Remove immediately and place in oil bath at 350-375 degrees Fahrenheit to temper.
9. Grind.
10. Inspect for hardness.

The crankshaft and master rod are both made of 3,140 steel and have the same general heat treatment as shown by the following for the rod:

1. After forging, anneal.
2. Rough machine.
3. Heat to 1,480-1,500 degrees Fahrenheit and quench in oil.
4. Draw in a lead bath to secure 277 to 321 B.h.n.
5. Pickle.

Double notch Izod impact specimens are faced and heat treated with every part calling for these properties, the average of the two impact values on the bars must be as noted above.

This material and heat treatment calls for the following minimum properties:

Ultimate Strength	128,000 psi	
Elongation %	Brinell Hardness	Izod Impact
17	277	56
16	286	52
15	293	48
14	302	45
13	311	42
12	321	40

Nitralloy and Nitriding.—Leon Guillet, a scientist, has presented to the French Academy of Sciences a description of a recently perfected "Nitrided steels" which he believes will revolutionize airplane manufacture. "At present the cylinders of airplane motors are composed of nickel and other alloys which wear out quickly," M. Guillet says. "As they wear the oil consumption mounts." Steels for nitriding are now being manufactured in the United States by Central Alloy Steel Corporation of Massillon, Ohio, and The Ludlum Steel Co. of Watervliet, N. Y., using the trade name "Nitralloy." For many years both producers and consumers have sought steels and case-hardening methods which would produce the ideal combination of extreme hardness, wear resistance and toughness without the difficulties, such as deformation, distortion and breakage, which frequently attend the carburizing process of hardening.

The nitriding process subjects special Nitralloy steels to the action of ammonia gas, for a period of time which varies with the depth of case desired, at a temperature of approximately 950 degrees Fahrenheit, without subsequent quenching. The case of nitrided Nitralloy will scratch glass with ease. Even special testing files wear smooth without effect on its extremely hard surface. Such a degree of hardness is beyond the effective range of the usual methods of hardness testing. Herbert Pendulum results read in terms of Brinell numerals show as high as 900-1,100 Brinell.

The manufacturing process employed can be summarized as follows:—The cylinder is first rough machined to within two to three millimeters of finished dimensions. In this state the part is given a quench and draw which will show a tensile strength of 90 to 95 kilograms per square millimeter. After this treatment, the outside is nearly finished leaving, however, a certain amount of material which should be removed after hardening. The bore is brought to finish dimensions, leaving only $\frac{1}{10}$ millimeter for straightening after hardening. In this state the outside is protected against nitriding so as to be able to finish machine without grinding after the hardening. The parts are then nitrided to a depth of $\frac{6}{10}$ to $\frac{7}{10}$ of a millimeter. After hardening, the parts are again machined. The protected outside is first finish machined, then the bore is rectified. The cylinders are then put in the blocks. This procedure yields cylinders of extreme hardness which cannot be filed and are perfectly interchangeable.

Nitriding insures that they will keep their very great hardness, even after the heating which they must undergo during normal running of the motor. The ordinary temperature reached in these cylinders is, however, quite inferior to that permitted by these nitrided steels whose hardness does not change up to 930 degrees Fahrenheit. Cylinders nitrided in this

way work equally well with cast iron or aluminum pistons. Pistons and rings are perfectly polished in contact with the hard surface of the cylinder and wear disappears nearly totally. The hardness of nitrided special steels can produce a very remarkable polish which changes entirely friction conditions. Aluminum alloys can rub directly on steels nitrided and polished without fear of wear or seizing. Motors are actually running with crankshafts of special nitrided steel without bronze or babbit bearings. Similarly nitrided and polished piston pins can be mounted directly in duralumin connecting rods. In recent tests, substituting the direct mounting of connecting rods on a nitrided crankshaft showed a gain of ten per cent in power at 3,000 r.p.m. and the maximum speed was increased to 4,000 r.p.m.

S. A. E. Non-Ferrous Metal Specifications.—In addition to the wide use that is made of alloy steel in the construction of parts of aviation engines, there are a number of non-ferrous metals that enter into the constructions of main and minor structural elements. The following specifications and general information should be of value to the reader.

WHITE BEARING METALS

The limits for the chemical compositions specified for metal in ingot form are closer than the limits specified for cast products, as allowances have been made for variations in the chemical content due to casting.

SPECIFICATION NO. 10, BABBITT

Composition in percentage:

	Cast Products	Ingots
Tin, min.	90	90.75
Copper	4 to 5	4.25 to 4.75
Antimony	4 to 5	4.25 to 4.75
Lead, max.	0.35	0.35
Iron, max.	0.08	0.08
Arsenic, max.	0.10	0.10
Bismuth, max.	0.08	0.08
Zinc and Aluminum	None	None

When finished bronze-backed bearings are purchased a maximum of 0.6 per cent lead is permissible in scraped samples provided a lead-tin solder has been used in bonding the bronze and the babbitt.

General Information.—This babbitt is very fluid and may be used for bronze-backed bearings, particularly for thin linings such as are used in aircraft engines. It is also suitable for die castings.

SPECIFICATION NO. 11, BABBITT

Composition in percentage:

	Cast Products	Ingots
Tin, min.	86.00	87.25
Copper	5.00 to 6.50	5.50 to 6.00
Antimony	6.00 to 7.50	6.50 to 7.00
Lead, max.	0.35	0.35
Iron, max.	0.08	0.08
Arsenic, max.	0.10	0.10
Bismuth, max.	0.08	0.08
Zinc and Aluminum	None	None

When finished bronze-backed bearings are purchased a maximum of 0.6 per cent lead is permissible in scraped samples provided a lead-tin solder has been used in bonding the bronze and the babbitt.

General Information.—This is a rather hard babbitt which may be used for lining connecting-rod and shaft bearings which are subjected to heavy pressures; its "wiping" tendency is very slight. It is also suitable for die castings.

SPECIFICATION NO. 12, BABBITT

Composition in percentage:

	Cast Products	Ingots
Antimony	9.50 to 11.50	10.25 to 10.75
Copper	2.25 to 3.75	2.75 to 3.25
Lead, max.	26.00	25.25
Tin, min.	59.50	60.00
Iron, max.	0.08	0.08
Bismuth, max.	0.08	0.08
Zinc and Aluminum	None	None

General Information.—This is a relatively cheap babbitt and is intended for bearings subjected to moderate pressures. It is also suitable for die castings.

SPECIFICATION NO. 13, BABBITT

Composition in percentage:

	Cast Products	Ingots
Tin	4.50 to 5.50	4.75 to 5.25
Antimony	9.25 to 10.75	9.75 to 10.25
Lead, max.	86.00	85.50
Copper, max.	0.50	0.50
Arsenic, max.	0.20	0.20
Zinc and Aluminum	None	None

General Information.—This is a cheap babbitt and serves successfully where the bearings are large and the service light. It should not be used as a substitute for a babbitt with a high tin content. It is also suitable for die castings.

SPECIFICATION NO. 14, BABBITT

Composition in percentage:

	Cast Products	Ingots
Tin	9.25 to 10.75	9.75 to 10.25
Antimony	14.00 to 16.00	14.75 to 15.25
Lead, max.	76.00	75.25
Copper	0.50	0.50
Arsenic, max.	0.20	0.20
Zinc and Aluminum	None	None

General Information.—This is a cheap babbitt and serves successfully where the bearings are large and the service light. It should not be used as a substitute for a babbitt with a high tin content. It is suitable for die castings.

ALUMINUM ALLOYS

SPECIFICATION NO. 30

Composition in percentage:

Aluminum, min.	90.00
Copper	7.00 to 8.50
Zinc, max.	0.20
Silicon, Iron, Zinc, Manganese and Tin, max.	1.70
Other Impurities	None

General Information.—The tensile strength of test-specimens about ½ inch diameter of this alloy cast in sand and tested without machining off the skin should be about 18,000 to 20,000 pounds per square inch and the elongation one to two per cent in two inches.

This is a light alloy having a specific gravity of about 2.83 and is used more extensively in the automotive industry than all other light casting alloys combined. A shrinkage of 0.156 ($\frac{5}{32}$) inch per foot should be allowed in pattern designs. This alloy is used for crankcases, oil-pans, steering-wheel spiders, differential carriers, transmission cases, camshaft housings, hub-caps and similar parts.

SPECIFICATION NO. 31

Composition in percentage:

Aluminum, min.	81.00
Copper	2.25 to 3.25
Zinc	12.50 to 14.50
Silicon, Iron, Manganese and Tin, max.	1.70
Other Impurities	None

General Information.—The tensile strength of test-specimens about ½ inch diameter of this alloy cast in sand and tested without machining off the skin should be about 25,000 to 30,000 pounds per square inch with an elongation of more than one per cent in two inches.

The specific gravity is about 3.0 and a shrinkage of 0.156 ($\frac{5}{32}$) inch per foot should be allowed in pattern designs.

This alloy is used extensively in England for such parts as crankcases, oil-pans, steering-wheel spiders and transmission cases.

SPECIFICATION NO. 32

Composition in percentage:

Aluminum, min.	85.50
Copper	11.00 to 13.50
Zinc, max.	0.20
Silicon, Iron, Zinc, Manganese and Tin, max.	1.70
Other Impurities	None

General Information.—The tensile strength of test-specimens about ½ inch diameter of this alloy cast in sand and tested without machining off

the skin should be about 19,000 to 23,000 pounds per square inch and the elongation will be practically nothing.

The specific gravity of this alloy is about 2.95 and a shrinkage of 0.156 (5/32) inch per foot should be allowed in pattern designs. This alloy is used for manifolds, pumps, carburetors, cylinders and other parts which should be free from leaks and where the brittleness of the alloy is not objectionable.

SPECIFICATION NO. 33

Composition in percentage:

Aluminum	88.00 to 92.00
Copper	6.00 to 8.00
Zinc, max.	2.50
Iron, max.	1.50
Silicon, Manganese and Tin, max.	0.75
Other Impurities	None

General Information.—The tensile strength of test-specimens about ½ inch diameter of this alloy cast in sand and tested without machining off the skin should be about 19,000 to 21,000 pounds per square inch with an elongation of one to two per cent in two inches.

This is a light alloy having a specific gravity of 2.83 to 2.86 and is used extensively in the automotive industry A shrinkage of 0.156 (5/32) inch per foot should be allowed in pattern designs.

This alloy is similar to Specification No. 30 and is used for crankcases, oil-pans, steering-wheel spiders, differential carriers, transmission cases, camshaft housings, hub-caps and similar parts.

SPECIFICATION NO. 34

Composition in percentage:

Aluminum, min.	87.00
Copper	9.25 to 10.75
Iron	0.90 to 1.50
Magnesium	0.15 to 0.35
All other elements, not over	0.75

General Information.—Test-bars cast in a chill mould show a tensile-strength of 24,000 to 30,000 pounds per square inch and the elongation in two inches is usually less than one per cent. The specific gravity should not exceed 2.95. The Brinell hardness number, using a 500 or 1,000-kilogram load with a ball ten millimeters in diameter, should be not less than 85 and should average about 105.

This alloy cast in permanent moulds is used principally for pistons, camshaft bearings, valve tappet-guides and other parts where high hardness and good bearing qualities are essential.

SPECIFICATION NO. 35, ALUMINUM

Composition in percentage:

Aluminum, min.	92.50
Copper, max.	0.60
Iron, max.	1.00
Silicon	4.50 to 6.50
Zinc, max.	0.20
Manganese, max.	0.20

General Information.—The minimum tensile-strength of test-specimens about ½ inch in diameter of this alloy cast in sand and tested without machining off the skin, should be about 16,000 pounds per square inch with a minimum elongation of 3.5 per cent in two inches.

This alloy is intended for automobile body parts and other parts that must be cast in thin sections. The alloy withstands salt-water corrosion very well and is therefore suitable for aircraft engine parts or other parts that may be subjected to severe corroding influences. The alloy has a relatively low yield-point and therefore cannot be used where great strength or stiffness is required.

CAST BRASS ALLOYS

SPECIFICATION NO. 40, RED BRASS

Composition in percentage:

Copper	83.00 to 86.00
Tin	4.50 to 5.50
Lead	4.50 to 5.50
Zinc	4.50 to 5.50
Iron, max.	0.35
Antimony, max.	0.25
Aluminum	None

General Information.—Good castings made of this alloy should give the following minima in physical characteristics:

Ultimate strength, lb. per sq. in.	27,000
Yield point, lb. per sq. in.	12,000
Elongation in 2 in. or proportionate gauge length, per cent	16

This is a free-cutting brass with good casting and finishing properties.

SPECIFICATION NO. 41, YELLOW BRASS

Composition in percentage:

Copper	62.00 to 65.00
Lead	2 00 to 4.00
Zinc	31.00 to 36.00
Tin, max.	1.00
Iron, max.	0.50
Aluminum	None
Other Impurities	0.25

General Information.—Good castings made of this alloy should give the following minima in physical characteristics:

Ultimate strength, lb. per sq. in.	25,000
Yield point, lb. per sq. in.	12,000
Elongation in 2 in. or proportionate gauge length, per cent	20

This alloy is intended for use in commercial castings where cheapness and good machining properties are the main considerations.

SPECIFICATION NO. 42, WHITE NICKEL BRASS

Composition in percentage:

Copper	55.00 to 64.00
Nickel, min.	18.00
Iron, max.	0.35
Aluminum	None
Other Impurities	0.25
Zinc	Remainder

General Information.—Good castings made of this alloy should give the following minima in physical characteristics:

Ultimate strength, lb. per sq. in.	30,000
Elongation in 2 in or proportionate gauge length, per cent	20

This brass is intended for use for trimmings or other parts requiring a metallic-white finish. The higher the nickel content, the more permanent will be the color.

SPECIFICATION NO. 43, MANGANESE BRONZE

This specification is substantially the same as Specification No. B7-14 of the American Society for Testing Materials.

Composition in percentage:

Copper	53.00 to 62.00
Zinc	38.00 to 47.00
Lead, max.	0.15

This metal may be hardened by the addition of small amounts of tin, iron, manganese, aluminum or a combination of these metals. Good sand castings made of this alloy should give the following minima in physical characteristics:

Tensile-strength, lb. per sq. in.	60,000
Yield-point, lb. per sq. in.	30,000
Elongation in 2 in. or proportionate gauge length, per cent	15

For the purpose of inspection, the most importance should be placed on the following minima in physical requirements on a special chill cast S. A. E. Standard test-bar. This test-bar shall be cut from one corner, near the bottom of the test ingot, cast in a properly tapered iron mould, approximately three inches deep by 2¾ inches wide by twelve inches long.

Tensile-strength, lb. per sq. in.	70,000
Elongation in 2 in., per cent	20

General Information.—This alloy is intended for use in castings where strength and toughness are required. It is equivalent to the copper-zinc alloys commercially known as Cast Manganese Bronze or its equivalents, such as Cast Tobin Bronze and Cast Naval Bronze.

SPECIFICATION NO. 44, CAST BRASS TO BE BRAZED

Composition in percentage:

Copper	83.00 to 86.00
Zinc	14.00 to 17.00
Lead, max.	0.50
Iron, max.	0.15

General Information.—This brass starts to melt at approximately 1,830 degrees Fahrenheit and is entirely melted at approximately 1,870 degrees Fahrenheit. As a brazing material on this brass to be brazed either Silver Solder or a brazing brass melting at a temperature lower than the brass to be brazed should be used.

SPECIFICATION NO. 45 BRAZING SOLDER

Composition in percentage:

Copper	48.00 to 52.00
Lead, max.	0.50
Iron, max.	0.10
Zinc	Remainder

General Information.—This solder starts to melt at approximately 1,560 degrees Fahrenheit and is entirely melted at approximately 1,600 degrees Fahrenheit. It may be used by melting it in a crucible under a flux of borax, with or without the addition of boric acid, and dipping the material to be brazed in the melted brazing solder; or this brazing solder, in a powdered form, may be mixed with the flux applied to the material to be brazed, and melted either in a furnace or by the use of a brazing torch.

BRONZE ALLOYS

Bearings or gears made of bronze alloys should be used only against hardened steel.

SPECIFICATION NO. 62, HARD CAST BRONZE

Composition in percentage:

Copper	86.00 to 89.00
Tin	9.00 to 11.00
Lead, max.	0.20
Iron, max.	0.06
Zinc	1.00 to 3.00

General Information.—Good castings made of this alloy should give the following minima in physical characteristics:

Ultimate strength, lb. per sq. in.	30,000
Yield point, lb. per sq. in.	15,000
Elongation in 2 in. or proportionate gauge length, per cent	14

This alloy is suitable wherever a strong general utility bronze is re-

quired. It may be used for severe working conditions where heavy pressures obtain, as in gears and bearings.

SPECIFICATION NO. 63, LEADED GUN METAL

Composition in percentage:

Copper	86.00 to 89.00
Tin	9.00 to 11.00
Phosphorous, max.	0.25
Zinc and other Impurities, max.	0.50
Lead	1.00 to 2.50

General Information.—Good castings made of this alloy should give the following minima in physical characteristics:

Ultimate strength, lb. per sq. in.	30,000
Yield point, lb. per sq. in.	12,000
Elongation in 2 in. or proportionate gauge length, per cent	10

Combining strength with fair machining qualities, this general utility bronze is especially good for bushings subjected to heavy loads and severe working conditions.

SPECIFICATION NO. 64, PHOSPHOR BRONZE

Composition in percentage:

Copper	78.50 to 81.50
Tin	9.00 to 11.00
Lead	9.00 to 11.00
Phosphorus	0.05 to 0.25
Zinc, max.	0.75
Other Impurities, max.	0.25

General Information.—Good castings made of this alloy should give the following minima in physical characteristics:

Ultimate strength, lb. per sq. in.	25,000
Yield point, lb. per sq. in.	12,000
Elongation in 2 in. or proportionate gauge length, per cent	8

This metal is an excellent composition for use where anti-friction qualities are desired, standing up exceedingly well under heavy loads and severe usage.

SPECIFICATION NO. 65, PHOSPHOR GEAR BRONZE

Composition in percentage:

Copper	88.00 to 90.00
Tin	10.00 to 12.00
Phosphorus	0.10 to 0.30
Lead, Zinc and other Impurities, max.	0.50

General Information.—Good castings made of this alloy should give the following minima in physical characteristics:

Ultimate strength, lb. per sq. in.	35,000
Yield point, lb. per sq. in.	20,000
Elongation in 2 in. or proportionate gauge length, per cent	10

This is a very hard bronze and may be used for gears and worm wheels where the requirements are severe.

SPECIFICATION NO. 66, BRONZE BACKING FOR LINED BEARINGS

Composition in percentage:

Copper	83.00 to 86.00
Tin	4.50 to 6.00
Lead	8.00 to 10.00
Zinc, max.	2.00
Impurities, max.	0.25

General Information.—Good castings made of this alloy should give the following minima in physical characteristics:

Ultimate strength, lb. per sq. in.	25,000
Yield point, lb. per sq. in.	12,000
Elongation in 2 in. or proportionate gauge length, per cent	8

This composition is recommended as an inexpensive but suitable alloy for bronze-backed bearings.

SPECIFICATION NO. 67, SEMI-PLASTIC BRONZE

Composition in percentage:

Copper	75.50 to 78.50
Tin	7.25 to 8.75
Lead	13.50 to 16.50
Zinc, max.	0.50
Phosphorus, max.	0.25
Iron, max.	0.25
Antimony, max.	0.50
Aluminum	None
Impurities, max.	0.75

General Information.—Good castings made of this alloy should give the following minima in physical characteristics:

Ultimate strength, lb. per sq. in.	20,000
Elongation in 2 in. or proportionate gauge length, per cent	10

This metal is intended for use where a soft bronze with good anti-friction qualities is desired.

SPECIFICATION NO. 68, CAST ALUMINUM BRONZE

Composition in percentage:

Copper	85.00 to 87.00

Aluminum	7.00 to 9.00
Iron	2.50 to 4.50
Tin (none desired), max.	0.50
Other Impurities, max.	0.25

General Information.—Good castings made of this alloy should give the following minima in mechanical properties:

Ultimate strength, lb. per sq. in.	65,000
Yield point, lb. per sq. in.	20,000
Elongation in 2 in. or proportionate gauge length, per cent	20

This is a non-corrodible alloy of great strength with a hardness equal to that of manganese bronze and good bearing qualities. It is suitable for use in worm wheels, gears and similar parts.

The adoption of S. A. E. Standards and Recommended Practices by the Society does not insure that users of constructions that incorporate such standards or recommended practices will not be liable for infringement of patents that may exist or may hereafter be issued, and does not constitute a recommendation of any patented or proprietary rights that may be involved.

Cylinder and Crankcase Retention Bolts.—Bolts, with which may be included other types of screw fastening, such as studs and set-screws, are frequently applied without proper consideration of the problems involved, writes A. C. Burgoine in the *Automobile Engineer*. Motor designers seldom make calculations of the size and number of bolts required for many purposes, relying upon experience in making a suitable selection. To check the sizes of important bolts is, of course, customary but even this is sometimes done in a rather perfunctory manner. As a consequence, wide differences are found in the practice of various motor builders, and troubles frequently arise in service. Insufficient attention is paid to the selection of material and to workmanship, and only too often bolts are purchased from outside sources of supply almost entirely on a basis of price. Even where bolts are made in the factory, the tendency is to use the cheaper qualities of steel and to cut the machining times until both accuracy and finish suffer. The usual material for bolts and other screw fastenings is low-carbon steel, commonly known as mild steel, but alloy steels are frequently used by manufacturers of aviation engines in the more important applications. Plain carbon-steel of rather higher carbon content also is used for bolts but in the heat-treated condition.

Even for relatively unimportant applications, cheap steels are a poor investment. Considering the ever-present risk of failure of low-grade materials, and in view of the low torsional strength of this class of steel, a very high factor of safety must be applied. Indeed, for general purposes, a factor of ten is not too high, this corresponding to a working stress of about 4,500 pounds for a twenty-ton steel. As a matter of fact, no conscientious engine designer should entertain the use of these cheap steels for any purpose whatsoever. Steel having a minimum ultimate-strength of 26 tons may be worked at a nominal factor of safety of eight with a working stress of about 7,000 pounds. For ordinary purposes this steel may prove fairly satisfac-

tory but its relatively low strength still necessitates the use of fairly large bolts for any given load, and in the smaller sizes great care is required to obviate the twisting off of threads by the careless use of the wrench. A very generally used steel for screwed parts in the better class of automobile work is a plain carbon-steel giving the minimum tensile-strength of 35 tons and having much better torsional strength than the milder steels. Owing to its greater uniformity and the better finish of the screw-threads, this steel may be worked at a lower factor of safety. For average work, a value of six may be used, corresponding to a stress of about 14,000 pounds, but naturally the designer must exercise judgment.

Alloy Steel Bolts.—Where still greater strength and reliability are required and particularly where the bolts are likely to be subjected to shock loads, two alternatives are available. The first is to use an alloy steel, such as three-per cent nickel steel which can be relied upon to give 45-tons ultimate tensile-strength with a yield-point not below 32 tons. This steel is rather expensive and does not machine as well as plain carbon-steel. It is not particularly hard on the surface and is a little likely to "pick-up" where any fretting occurs, or when driven into a tight reamed hole. With a good resistance to shock and the minimum Izod-value of 40 feet-pounds, nickel steel may be worked with a nominal factor of safety of five, at which the stress will be about 20,000 pounds per square inch.

For bolts that are to be yet more highly stressed, the ternary alloy-steels are essential, but these find few applications in the ordinary automobile. In racing cars and aircraft engines, where weight must be cut and cost is unimportant, they are widely used, but nickel-chrome steel bolts cost two or three times as much as those of good mild steel and therefore are out of the question in the case of low-priced automotive products. Owing to the difficulties attendant upon the manufacturers of these steels, very careful inspection at all stages is essential, and heat-treatment must be most carefully carried out after machining, the bolts being finally finished by grinding to diameter. Nickel-chrome steel should give an ultimate tensile-strength of at least 55 tons, with the minimum yield-point of 45 tons. By suitable tempering the strength may be increased but at the expense of the impact-test value, which must be the minimum of 40 feet-pounds on a standard test-piece. For a 60-ton steel the Izod figure may be 50 feet-pounds.

Speaking generally, the adoption of bolts made from these high-tensile steels will not permit of any reduction in the size of bosses, lugs and flanges, because sufficient area must be provided to withstand the local crushing loads. Where the bolts pass through aluminum, increasing the sections and providing large heads and washers under the nuts may even be necessary. Any saving in weight due to the use of the high-tensile steel is thus largely illusory, and such bolts can generally be usefully employed only in conjunction with parts of steel, such as connecting rods.

Heat-Treated Carbon-Steel Bolts.—The second alternative where bolts are required to have strengths greater than ordinary mild steel is to use a plain carbon-steel of rather higher carbon-content and to heat-treat this by a process of hardening and tempering. Using a medium-carbon steel, by changing the heat-treatment the ultimate tensile-strength can be varied from 40 to as high as 65 tons, but as a general rule bolts are tempered to

between 40 and 50 tons ultimate-stress. With the lower ultimate-strength, the elongation is about 26 per cent and the reduction of area 65 per cent. At the same time the impact value in the Izod test is as high as 80 feet-pounds. As a general average, it may be accepted that the ultimate strength of the standard quality of bolts is 45 tons, with an Izod value of 70 feet-pounds.

From these figures it will be evident that the heat-treated carbon-steel bolts compare very favorably with those of nickel steel. Further, with the same average tensile-strength, the Izod value is decidedly higher, and thus the resistance to shock loads should be better. It is reasonable to work with the same factor of safety as for nickel steel, and therefore the working stress may be taken at 20,000 pounds for the 45-ton tempered bolts. Plain carbon-steel in the heat-treated condition is at least as reliable as the alloy steels, and the steel makers find much less difficulty in obtaining uniformity in quality and in test values. It machines more readily and to a better finish than alloy steel of the same strength, while dies and tools generally hold their cutting edges and dimensions better when used on carbon steel. Therefore, the expenses for tooling are reduced and there is less dimensional scrap. Heat-treatment is simple, and the risk of failure or of irretrievably ruining the material is less. The surface of a heat-treated carbon-steel bolt is harder than that of a bolt of alloy steel of equal strength, and this is a valuable feature. Where nuts are to be removed and replaced very frequently, the harder surface of the threads prevents the rapid wear that takes place with softer steels. Driven bolts do not so readily become scored and are stiffer to resist bending. They stand-up better to shear loads and are not so likely to become cut-up when the fit is not exactly tight. Heat-treated bolts successfully resist the torsional loads due to the spanner, and the fine finish obtainable on the threads reduces such loads to the minimum. Bolts of heat-treated steel may be tightened up closer than those of mild steel, even in a size smaller, and therefore are less likely to work loose. Owing to the harder surfaces, the threads do not wear appreciably, and after dismantling it will usually be found that the same relative positions of nut and bolt will enable the split pin or other locking device to be used as before. The nuts and bolt heads resist the attacks of wrenches very well, and the good appearance of the job is maintained.

Hardness Testing Methods.—Hardness, machining properties and wearing qualities are three totally different characteristics of a metal, and in selecting a method of hardness testing the other properties required in the finished component must be kept in mind. Among the numerous methods that have been adopted for testing hardness, perhaps the most usual shop-method consists merely in trying the effect of a small dead-smooth file, and after constant practice the degree of accuracy that can be obtained by this method alone is astonishing. It cannot, of course, be relied upon to give good comparative results with all metals, but it is useful for testing the suitability of chisels, punches, shear blades, and kindred tools. A similar test consists in trying the effect of the metal on ordinary glass. If it is possible to scratch the glass, the metal is in a "glass-hard condition," distinctly hard and probably very brittle.

Foremost among the more scientific methods of hardness testing, the Brinell method may be mentioned as being the one most universally adopted, in spite of its many disadvantages. As a means of determining the approximate tensile-strength of steel and thus saving the expense of machining a test piece, it is excellent. The hardness in this case is calculated from the area or width of the identation produced by applying an extremely hard steel ball with a given load to the polished surface of the metal. This is read by a special scale fitted with a microscopic eyepiece so very small indentations may be measured.

Usually the load is 3,000 kilograms, (6,613.87 pounds) when using a ball ten millimeters (0.3937 inches) in diameter on steel while loads of 500 to 1,000 kilograms (1,102.31 to 2,204.62 pounds) are used for nonferrous metals and alloys. In the latest machines hardened conical penetrators and diamond points of pyramid form are employed in place of the steel ball, but the method remains substantially the same.

The Brinell Test.—With regard to the limitations of the Brinell test, it is first of all obvious that components of light section cannot be adequately tested because the ball may cause distortion right through the metal. Extremely hard metals also cannot be satisfactorily tested since they are apt to distort the ball. In this connection the importance of frequently renewing the ball may be emphasized, since, should it become flattened, it would give very erroneous results. A further objection to the Brinell test is that the impression produced on hard steel is so small that it is frequently difficult to measure it with sufficient accuracy. To overcome this a ball which has previously been etched for about one minute in a two-per cent nitric acid in alcohol solution, may conveniently be substituted for the ordinary polished ball. This gives an impression with more clearly defined edges, so that the exact measurement is simplified.

Still another drawback to the Brinell test is that metals having a coarsely crystalline structure will give very different results according to the position of the indentation. Moreover, when the adjacent crystals are of varying hardness, accurate measurement is impossible. For similar reasons unreliable results will be obtaned from cast iron containing a large percentage of free soft graphite.

Difficulties are also likely to arise in testing case-hardened steel by the Brinell method, particularly if the depth of the case is not great. The case itself may consist of high-carbon steel having a Brinell hardness of over 600, while the low-carbon core may have a hardness number between 120 and 180. In such circumstances, if the case is at all thin, the deformation is likely to extend into the soft core, giving rise to entirely erroneous results. To detect defects in homogeneity of metals by the Brinell method is sometimes possible. It will also indicate heat-treatment when varying results are obtained from tests taken at a number of different points on a heat-treated article.

The Scleroscope Method.—The scleroscope test is also extremely easy to apply, while the small indentation that it produces will not mar even the lightest and most delicate component. The scleroscope hardness of hard steel is approximately 100. This test is useful for ascertaining the hardness of thin sections of metal and can also be employed for case-hard-

ened steel. In the scleroscope a small diamond pointed hammer is allowed to drop by a special trigger release from a nominal height and its rebound, after it hits the specimen tested is an indication of the hardness of the piece. Another type measures the indentation of a pyramid pointed member directly on a gauge, the point being pressed into the piece to be tested by a definite force or load. Obviously, however, the results obtained are influenced by different factors from those governing the Brinell test because, to quote an exaggerated case, substances such as rubber and celluloid give readings equal to those obtained with hardened steel. Were it not that the hammer produced a minute permanent indentation in the metal, this test would give a true indication of the elasticity of the specimen.

QUESTIONS FOR REVIEW

1. Describe why aircraft engines should have minimum vibration.
2. Outline two balancing devices for use with four cylinder engines.
3. What are the common methods of crankshaft construction?
4. When are counterbalanced crankshafts used in aircraft engines?
5. Are anti-friction bearings practical for supporting crankshafts of aircraft engines?
6. What is normalized steel?
7. What is the usual method of crankcase construction for Vee engines—for static radial engines?
8. Name common types of anti-friction bearings.
9. What material has the most strength for its weight?
10. What is Y alloy and where is it used?
11. Name some engine parts made of nickel steel.
12. What is the nitriding process?

CHAPTER XXIV

EARLY AND PRE-WAR AVIATION ENGINES

Aviation Engine Types—Early Anzani Engines—Anzani Y Engine—Anzani Connecting Rod Construction—Canton-Unné (Salmson) Engines—Stroke Equalizing Mechanism—Salmson Valve Timing—Construction of Early Gnome Motor—Cylinders Machined from Solid Bar—Exhaust Valve Mounting—Pistons Carry Inlet Valves—Exhaust Valve Operation—Why Odd Numbers of Cylinders Are Used—Gnome Carburetion and Lubrication—Rotary Motor Disadvantages—Gnome "Monosoupape" Type—Details of Cylinder Construction—Gnome "Monosoupape" Fuel System—Monosoupape Ignition—Monosoupape Lubrication—German "Gnome" Type Engine—The Le Rhone Rotary Motor—Le Rhone Connecting Rod Arrangement—Le Rhone Valve Actuation—Le Rhone Carburetor—Le Rhone Engine Action—Renault Air-Cooled Vee Engine.

Inasmuch as numerous forms of airplane engines have been devised, it would require a volume of considerable size to describe even the most important developments of recent years. As considerable explanatory matter has been given in preceding chapters and the principles involved in internal-combustion engine operation considered in detail, a review of the features of some of the most successful early airplane motors should suffice to give the reader a complete enough understanding of the art so all types of engines can be readily recognized and the advantages and disadvantages of each type understood, as well as defining the constructional features enough so the methods of locating and repairing the common engine and auxiliary system troubles will be fully grasped.

Aviation Engine Types.—Aviation engines can be divided into three main classes. One of the earliest attempts to devise distinctive powerplant designs for aircraft involved the construction of engines utilizing a radial arrangement of the cylinders or a star-wise disposition. Among the engines of this class may be mentioned the Anzani, R. E. P. and the Salmson or Canton and Unné forms. The two former are air cooled, the latter design has been made in both air-cooled and water-cooled. Engines of this type have been built in cylinder numbers ranging from three to twenty. While the simple forms were popular in the early days of aviation engine development, they have been succeeded by the more conventional arrangements which now form the largest class.

The reason for the adoption of a star-wise arrangement of cylinders has been previously considered. Smoothness of running can only be obtained by using a considerable number of cylinders. The fundamental reason for the adoption of the star-wise disposition is that a better distribution of stress is obtained by having all of the pistons acting on the same crankpin so that the crank-throw and pin are continuously under maximum stress. Some difficulty has been experienced in lubricating the lower cylinders in some forms of six-cylinder, rotary crank, radial engines but these have been largely overcome so they are not as serious in practice as a theoretical consideration would indicate.

Another class of engines developed to meet aviation requirements is a complete departure from the preceding class, though when the engines are

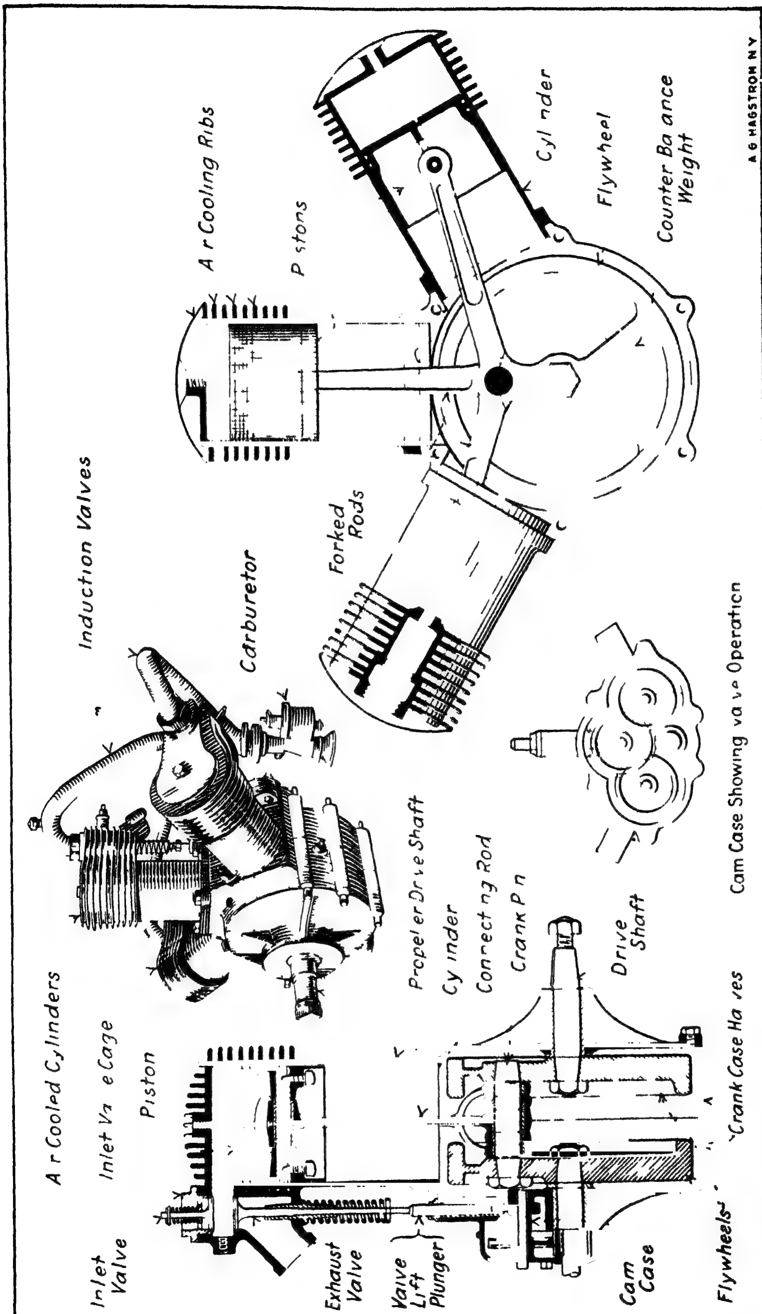


Fig. 365.—Views Outlining Construction of Early Three-Cylinder Anzani Aviation Motor, the First Engine to Drive an Airplane Across the English Channel, from France to England.

at rest, it is difficult to differentiate between them. This class includes engines having a star-wise disposition of the cylinders but the cylinders themselves and the crankcase rotate and the crankshaft remains stationary. The important rotary engines are the Gnome, the Le Rhone and the Clerget. The latter differed chiefly in the method of valve actuation, separate push rods being used for each valve.

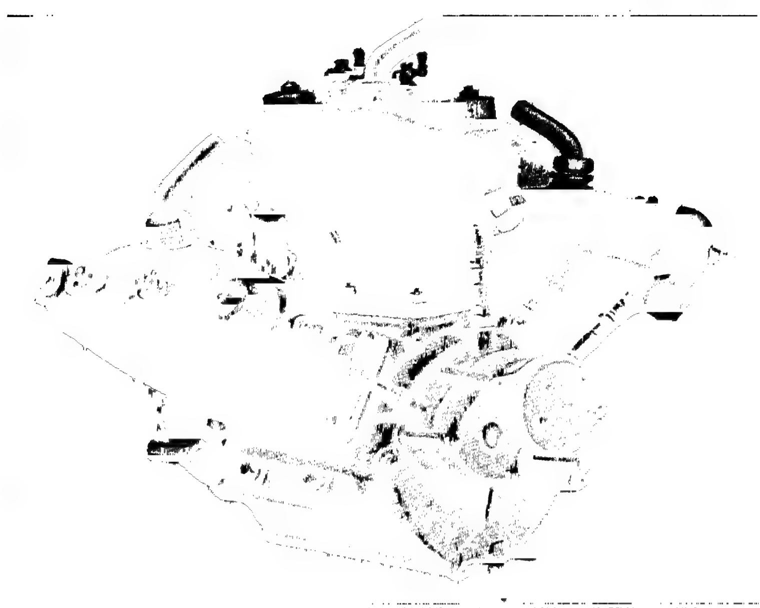


Fig. 366.—The Early Anzani "W" Type, Six-Cylinder Water-Cooled Aviation Engine.

By far the most important classification is that including engines which retain the approved design of the types of powerplants that have been so widely utilized in automobiles and which have modifications to increase reliability and mechanical strength and produce a reduction in weight. This class includes the vertical engines such as the Duesenberg and Hall-Scott four-cylinder; the Wisconsin, Aeromarine, Mercedes, Benz, and Hall-Scott six-cylinder vertical engines and the numerous eight- and twelve-cylinder Vee designs such as the Curtiss, Renault, Thomas-Morse, Sturtevant, Sunbeam, and others.

Early Anzani Engines.—The attention of the mechanical world was first directed to the great possibilities of mechanical flight when Bleriot crossed the English Channel in July, 1909, in a monoplane of his own design and construction, having the power furnished by a small three-cylinder air-cooled engine rated at about 24 horsepower and having cylinders 4.13 inches bore and 5.12 inches stroke, stated to develop the power at about 1,600 r.p.m. and weighing 145 pounds. The arrangement of this early Anzani engine is shown at Fig. 365, and it will be apparent that in the main, the lines worked out in motorcycle practice were followed to a large extent. The crankcase was of the usual vertically divided pattern,

the cylinders and heads being cast in one piece and held to the crankcase by stud bolts passing through substantial flanges at the cylinder base. In order to utilize but a single crankpin for the three cylinders it was necessary to use two forked rods and one rod of the conventional type. The arrangement shown at Fig. 365, called for the use of counter-balanced flywheels which were built up in connection with shafts and a crankpin to form what corresponds to the usual crankshaft assembly.

The inlet valves were of the automatic type so that a very simple valve mechanism consisting only of the exhaust valve push rods was provided.

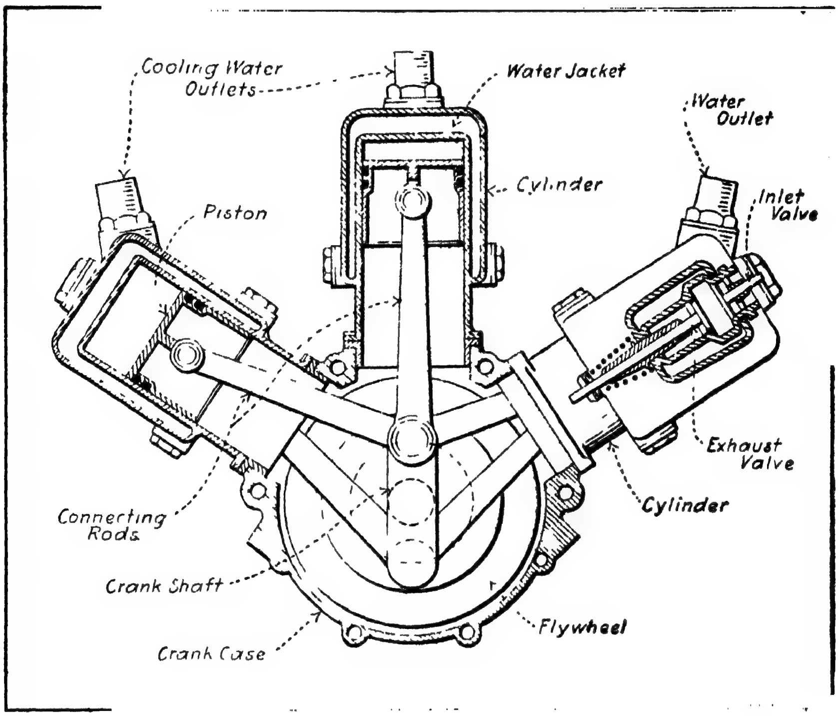


Fig. 367.—Transverse Sectional Elevation of the Anzani Six-Cylinder Water-Cooled Aviation Engine, which is the Ancestor of Modern Engines Using that Method of Cylinder Arrangement.

One of the difficulties of this arrangement of cylinders was that the impulses are not evenly spaced. For instance, in the forms where the cylinders were placed 60 degrees apart the space between the firing of the first cylinder and that next in order was 120 degrees crankshaft rotation, after which there was an interval of 300 degrees before the last cylinder to fire delivered its power stroke. In order to increase the power given by the simple three-cylinder air-cooled engine a six-cylinder water-cooled type, as shown at Figs. 366 and 367 was devised. This was practically the same in action as the three-cylinder except that a double throw crankshaft was used and while the explosions were not evenly spaced the number of explosions obtained resulted in fairly uniform application of power.

Anzani Y Engine.—The design of three-cylinder Anzani engine, which was used to some extent for school machines, is shown at Fig. 368. In this, the three cylinders are symmetrically arranged about the crankcase or 120 degrees apart. The balance is greatly improved by this arrangement and the power strokes occur at equal intervals of 240 degrees of crankshaft rotation. This method of construction is known as the Y design. By grouping two of these engines together, as outlined at Fig. 368, which gives an internal view, and at Fig. 369, which shows the sectional view, and

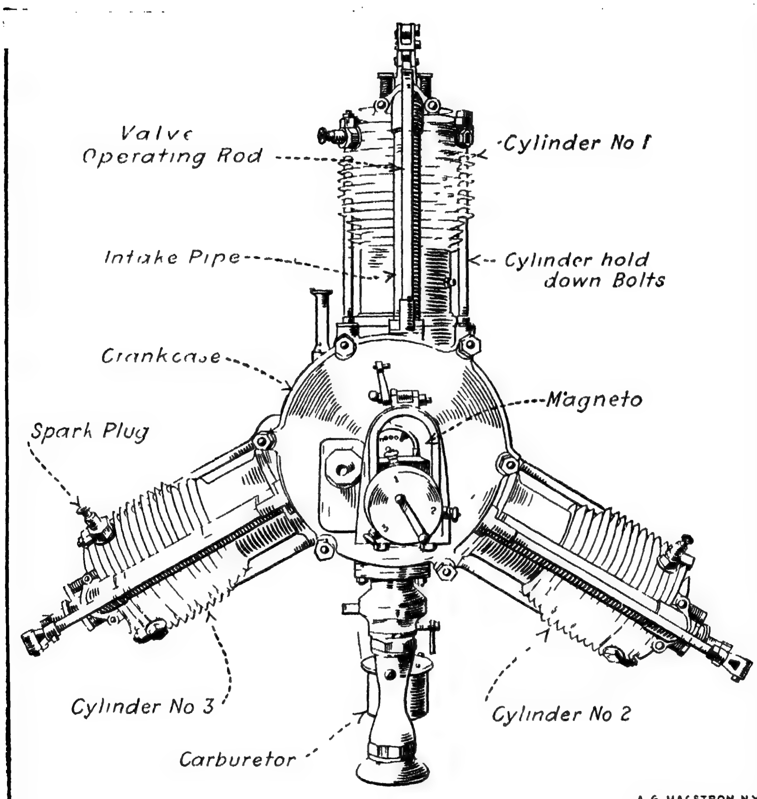


Fig. 368.—The Three-Cylinder Air-Cooled Anzani Aviation Engine was the Pioneer "Y" Form Powerplant, and One of the First Static Radial Motors.

using the ordinary form of double throw crankshaft with crankpins separated by 180 degrees, a six-cylinder radial engine is produced which runs very quietly and furnishes a steady output of power. The peculiarity of the construction of this engine is in the method of grouping the connecting rod about the common crankpin without using forked rods or the "mother rod" system employed in the Gnome engines.

Anzani Connecting Rod Construction.—In the Anzani the method followed is to provide each connecting rod big end with a shoe which consists of a portion of a hollow cylinder held against the crankpin by split clamping

rings. The dimensions of these shoes are so proportioned that the two adjacent connecting rods of a group of three will not come into contact even when the connecting rods are at the minimum relative angle. The three shoes of each group rest upon a bronze sleeve which is in halves and which surrounds the crankpin and rotates relatively to it once in each crankshaft revolution. The collars, which are of tough bronze, resist the inertia forces while the direct pressure of the explosions is transmitted directly to the crankpin bushing by the shoes at the big end of the connecting rod. The same method of construction, modified to some extent, is used in the LeRhone rotary cylinder engine.

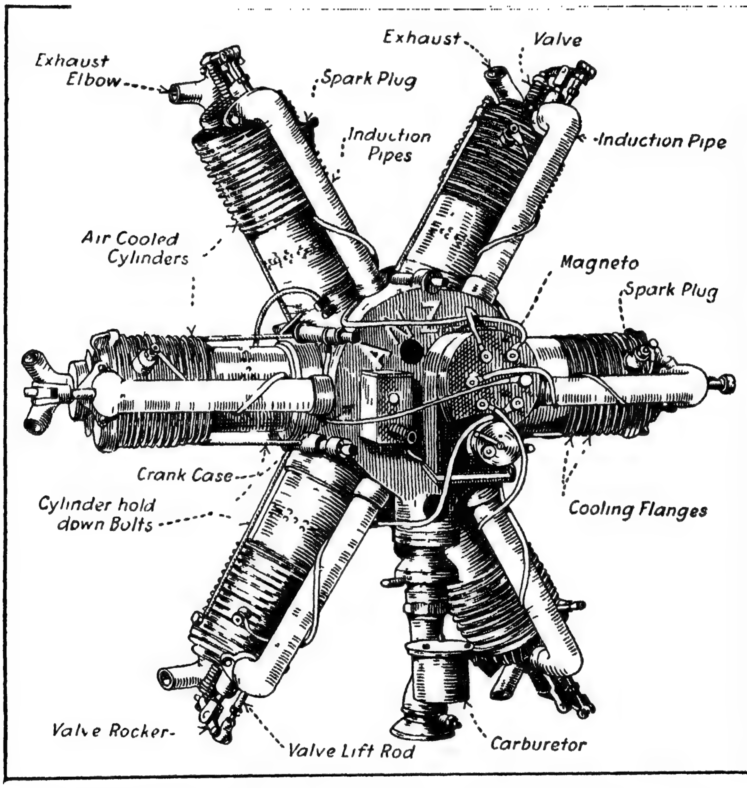


Fig. 369.—The Early Anzani Fixed Crankcase Motor of the Six-Cylinder Radial Form was One of the First of this Type to Utilize Air Cooling Successfully.

Both cylinders and pistons of the Anzani engines are of cast iron, the cylinders being provided with a liberal number of cooling flanges which are cast integrally. A series of auxiliary exhaust ports is drilled near the base of each cylinder in some models so that a portion of the exhaust gases will flow out of the cylinder when the piston reaches the end of its power stroke. This reduces the temperature of the gases passing around the

exhaust valves and prevents warping of these members. Another distinctive feature of this early engine design is the method of attaching the Zenith carburetor to an annular chamber surrounding the rear portion of the crankcase from which the intake pipes leading to the intake valves

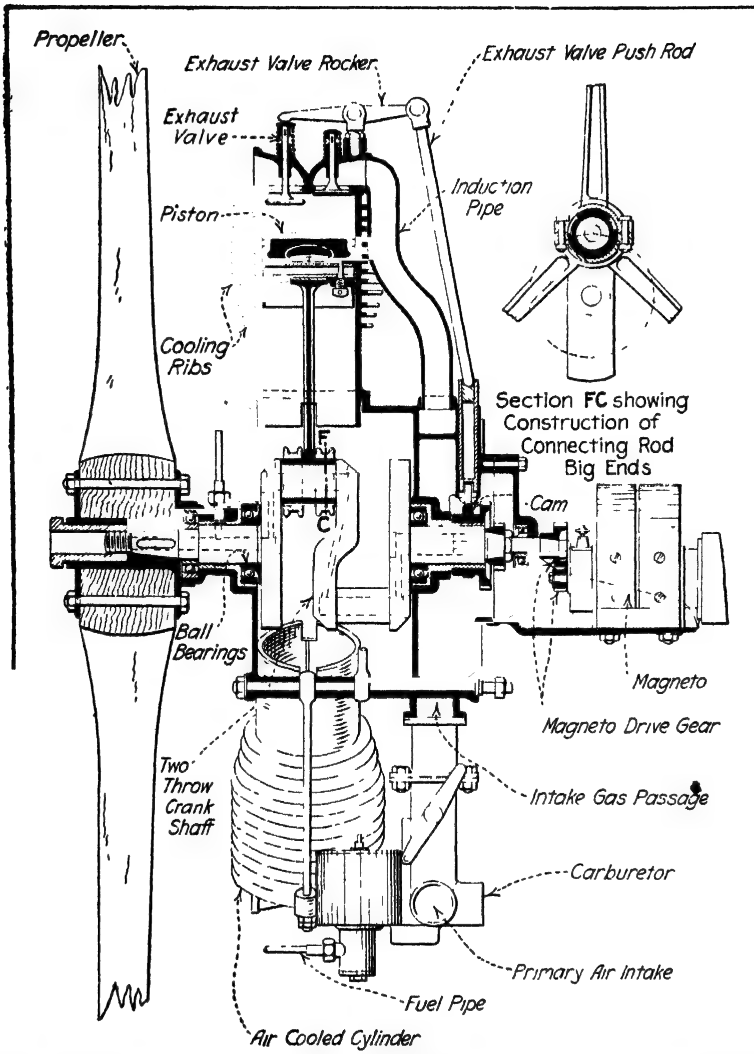


Fig. 370.—Sectional View Showing Internal Parts in Pioneer Form of Six-Cylinder Anzani Engine, which was so Successful that it Has Been Changed Only in Detail in More Recent Designs.

radiate. The magneto is the usual six-cylinder form having the armature geared to revolve at one and one-half times crankshaft speed.

The Anzani aviation engines are also made in ten- and twenty-cylinder forms as shown at Fig. 371. It will be apparent that in the ten-cylinder form explosions will occur every 72 degrees of crankshaft rotation, while

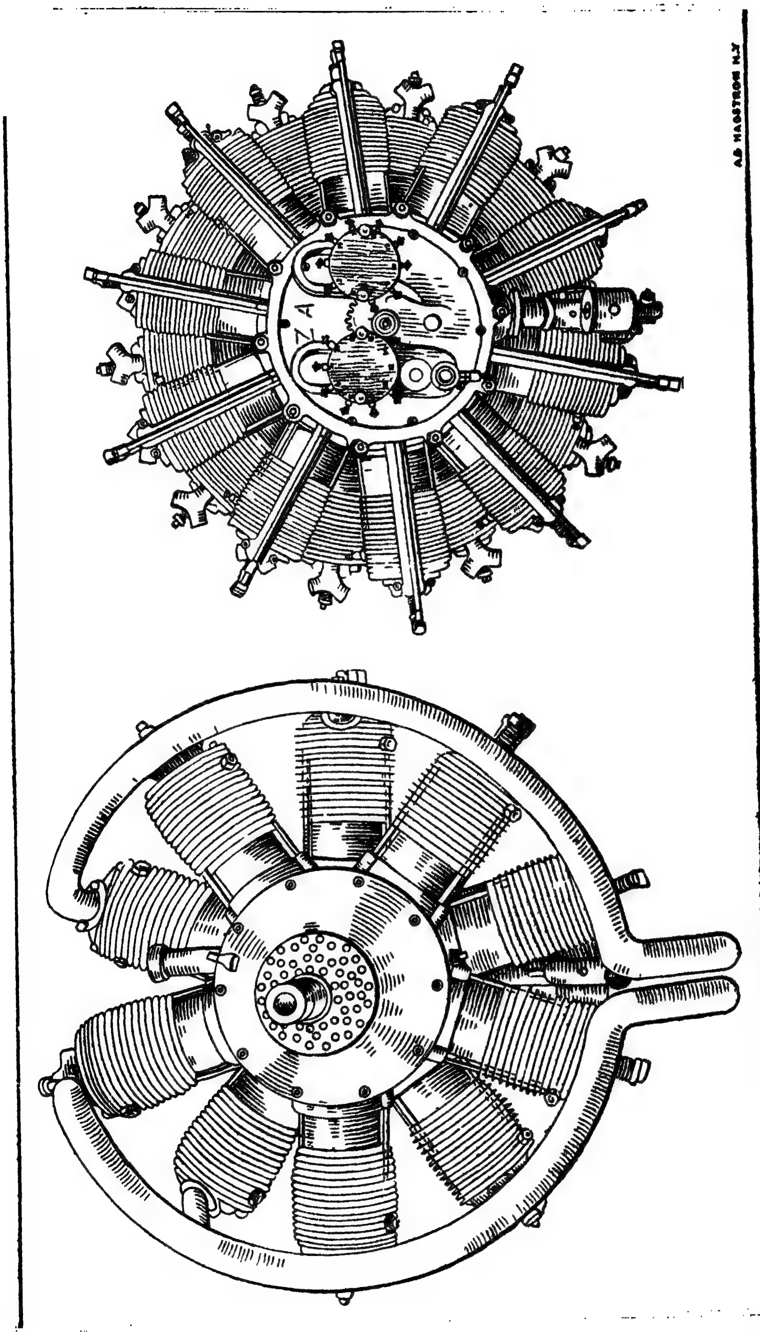


Fig. 371.—Drawings Showing Anzani Multiple-Cylinder Radial Engines of Early Design. Ten-Cylinder Static Radial is Shown at the Left and Twenty-Cylinder Fixed Type at the Right

in the twenty-cylinder, 200 horsepower engine at any instant five of the cylinders are always working and explosions are occurring every 36 degrees of crankshaft rotation. On the twenty-cylinder engine, two carburetors are used and two magnetos, which are driven at two and one-half times crankshaft speed. The general cylinder and valve construction is practically the same as in the simpler engines.

Fig. 372.—A Five-Cylinder Fan Shaped Air-Cooled Motor Was Designed by Robert Esnault Pelterie, and Applied to a Very Efficient Monoplane of His Own Design in the Earliest Days of Aviation.

Salmson—Canton and Unné Engines.—This engine, which has been devised specially for aviation service, is generally known as the "Salmson" and was manufactured in both France and Great Britain. It is a nine-cylinder water-cooled radial engine, the nine-cylinders being symmetrically disposed around the crankshaft while the nine connecting rods all operate on a common crankpin in somewhat the same manner as the rods in the

Gnome motor. The crankshaft of the Salmson engine is not a fixed one and inasmuch as the cylinders do not rotate about the crankshaft it is necessary for that member to revolve as in the conventional engine. The stout hollow steel crankshaft is in two pieces and has a single throw. The crankshaft is built up somewhat the same as that of the Gnome engine. Ball bearings are used throughout this engine as will be evident by inspecting the sectional view given at Fig. 374. The nine steel connecting rods are machined all over and are fitted at each end with bronze bushings, the distance between the bearing centers being about 3.25 times crank length.

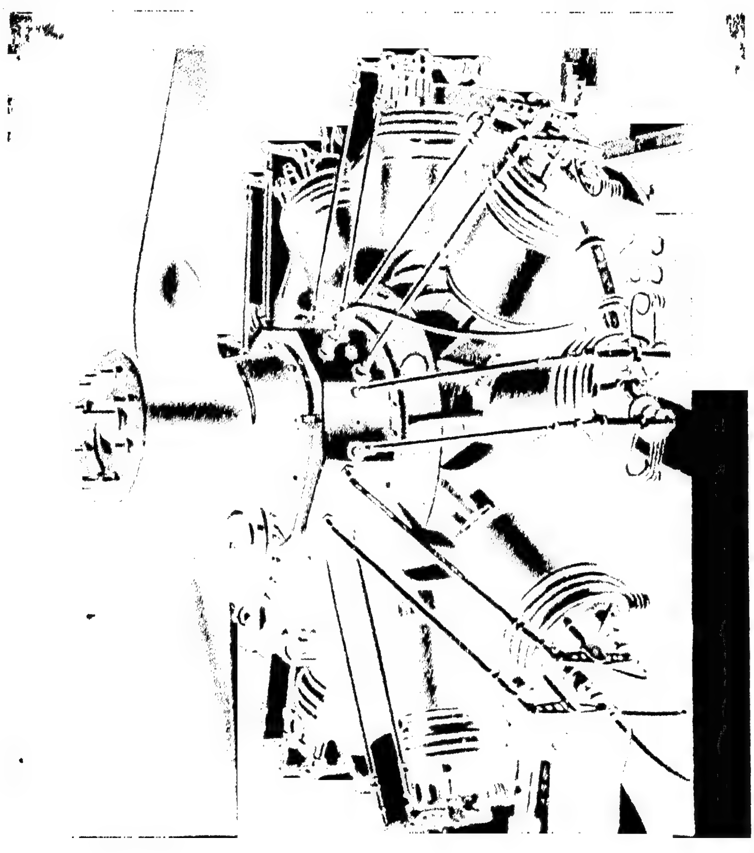


Fig. 373.—The Canton and Unné (Salmson) Nine-Cylinder Water-Cooled Radial Engine, which Gave Very Good Service During the War.

Stroke Equalizing Mechanism.—The method of connecting up the rods to the crankpin is one of the characteristic features of this design. No "mother" rod as supplied in the Gnome engine is used in this type inasmuch as the steel cage or connecting rod carrier is fitted with symmetrically disposed big end retaining pins. Inasmuch as the carrier is mounted on ball bearings some means must be provided of regulating the motion

of the carrier as if no means were provided the resulting motion of the pistons would be irregular.

The method by which the piston strokes are made to occur at precise intervals involves a somewhat lengthy and detailed technical explanation. It is sufficient to say that an epicyclic train of gears, one of which is rigidly attached to the crankcase so it cannot rotate is used, while other gears make a connection between the fixed gear and with another gear which is exactly the same size as the fixed gear attached to the crankcase and which is

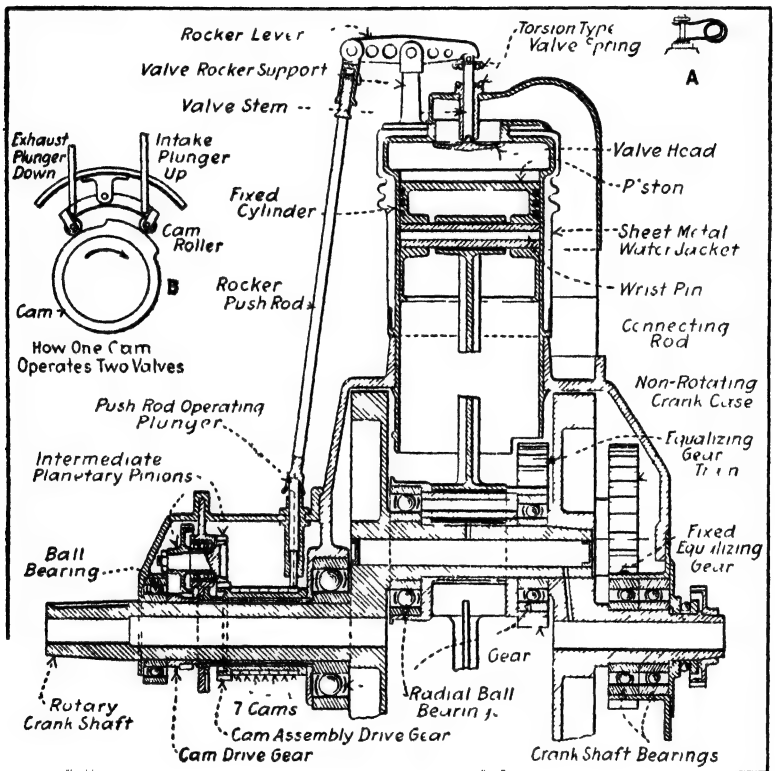


Fig. 374.—Sectional View Showing Construction of the Canton and Unné Water-Cooled Static Radial Cylinder Engine.

formed integrally with the connecting rod carrier. The action of the gearing is such that the cage carrying the big end retaining pins does not rotate independently of the crankshaft, though, of course, the crankshaft or rather crankpin bearings must turn inside of the big end carrier cage.

Cylinders of this engine are of nickel steel machined all over and carry water jackets of spun copper which are attached to the cylinders by brazing. The water jackets are corrugated to permit the cylinder to expand freely. The ignition is similar to that of the fixed crank rotating cylinder engine. An ordinary magneto of the two spark type driven at 1¾ times crankshaft

speed is sufficient to ignite the seven-cylinder form, while in the nine-cylinder engines the ignition magneto is of the "shield" type giving four sparks per revolution. The magneto is driven at 1⅑ times crankshaft speed. Nickel steel valves are used and are carried in castings or cages which screw into bosses in the cylinder head.

Salmson Valve Timing.—Each valve is cam operated through a tappet, push rod and rocker arm, seven cams being used on a seven-cylinder engine and nine cams on the nine-cylinder. One cam serves to open both valves as in its rotation it lifts the tappets in succession and so operates the exhaust and inlet valves respectively. This method of operation involves the same period of intake and exhaust. In normal engine practice the inlet valve opens twelve degrees late and closes twenty degrees late. The exhaust opens 45 degrees early and closes six degrees late. This means about 188 degrees in the case of inlet valves and 231 degrees crankshaft travel for exhaust valves. In the Salmson engine, the exhaust closes and the inlet opens at the outer dead center and the exhaust opens and the inlet closes at about the inner dead center. This engine is also made in a fourteen-cylinder 200 brake horsepower design which is composed of two groups of seven-cylinders, and it has been made in an eighteen-cylinder design of 600 horsepower. The nine-cylinder 130 horsepower has a cylinder bore of 4.73 inches and a stroke of 5.52 inches. Its normal speed of rotation is 1,250 r.p.m. The weight was 4¼ pounds per brake horsepower.

Construction of Early Gnome Motor.—It cannot be denied that for a time one of the most widely used of airplane motors was the seven-cylinder revolving air-cooled Gnome, made in France. For a total weight of 167 pounds this motor developed 45 to 47 horsepower at 1,000 revolutions, being equal to 3.35 pounds per horsepower, and had proved its reliability by securing many early long-distance and endurance records. The same engineers produced a nine-cylinder and by combining two single engines a fourteen-cylinder revolving Gnome, having a nominal rating of 100 horsepower, with which world's speed records were broken at the time of its introduction. A still more powerful engine has been made with eighteen-cylinders. The nine-cylinder "monosoupape" delivers 100 horsepower at 1,200 r.p.m., the engine of double that number of cylinders is rated at about 180 horsepower.

Except in the number of cylinders and a few mechanical details the fourteen-cylinder motor is identical with the seven-cylinder one; fully three-quarters of the parts used by the assemblers would do just as well for one motor as for the other. Owing to the greater power demands of the airplane the smaller sizes of Gnome engines were not used much in service airplanes except for school machines. There is very little in this motor that is common to the standard type of vertical motorcar engine. The cylinders are mounted radially round a circular crankcase; the crankshaft is fixed, and the entire mass of cylinders and crankcase revolves around it as outlined at Fig. 375. The explosive mixture and the lubricating oil are admitted through the fixed hollow crankshaft, passed into the explosion chamber through an automatic intake valve in the piston head in the early pattern, and the spent gases exhausted through a mechanically operated valve in the cylinder head. The course of the gases is practically

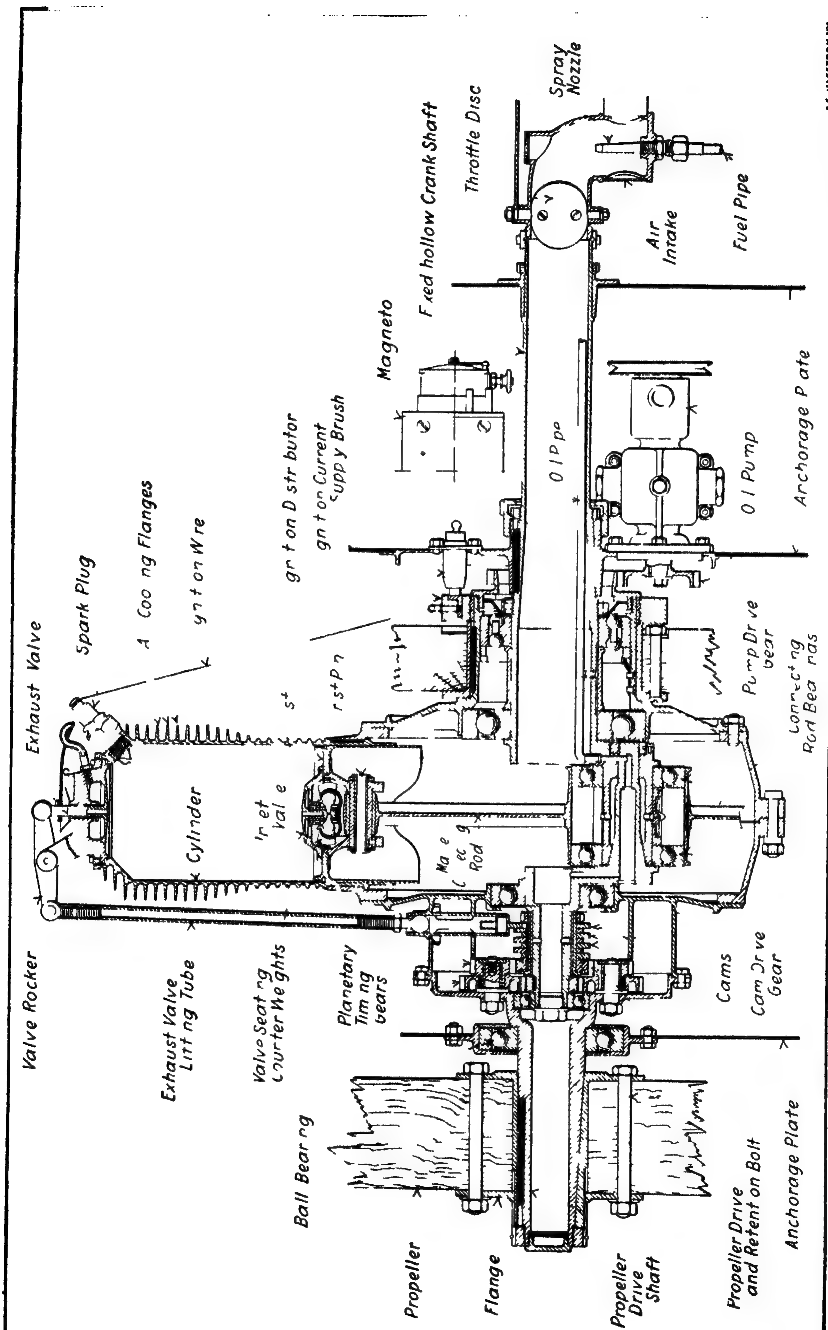


Fig. 375.—Longitudinal Sectional View Outlining Construction of Early Type Gnome Motor, Having Induction Valve in the Piston Top.

a radial one. A peculiarity of the construction of the motor is that nickel steel is used throughout. Aluminum is employed for the two oil pump housings; the single compression ring known as the "obdurator" for each piston is made of brass; there are three or four brass bushes; gun metal is employed for certain pins—the rest is machined out of chrome nickel steel.

The crankcase is practically a steel hoop, the depth depending on whether it has to receive seven- or fourteen-cylinders; it has seven or

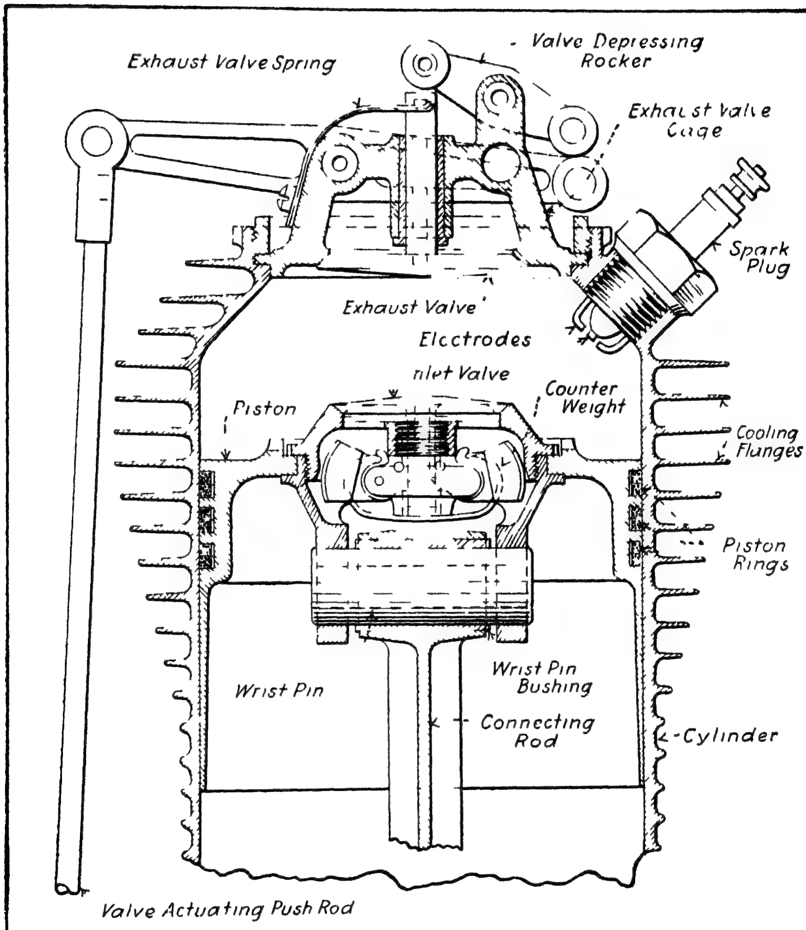


Fig. 376.—Sectional View of Early Type Gnome Cylinder and Piston, Showing Construction, Application, and Method of Operation of Inlet and Exhaust Valves.

fourteen holes bored as illustrated on its circumference. When fourteen or eighteen cylinders are used the holes are bored in two distinct planes, and offset in relation one to the other.

Cylinders Machined from Solid Bar.—The cylinders of the small engine which have a bore of 4 $\frac{3}{10}$ inches and a stroke of 4 $\frac{7}{10}$ inches, are

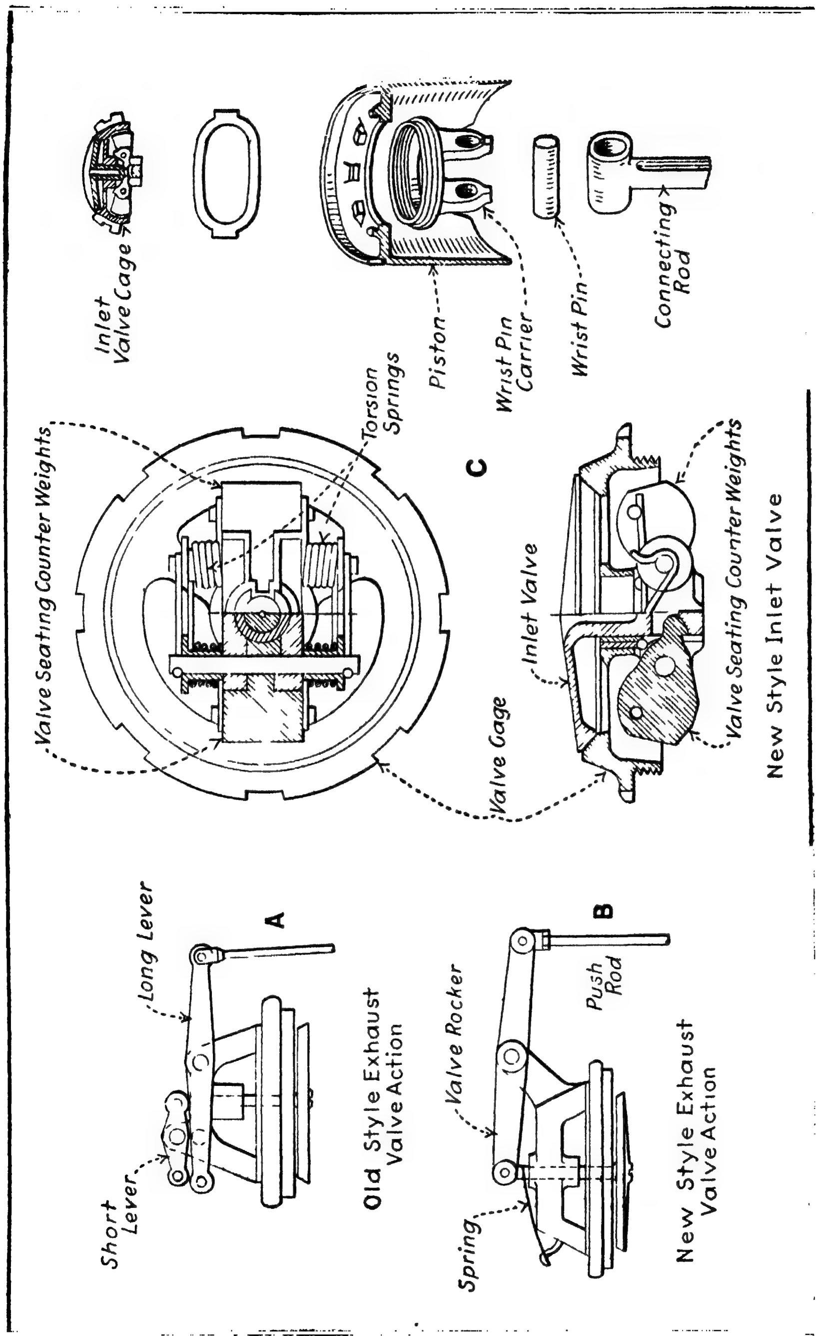


Fig. 377.—Details of Old Style Gnome Motor Inlet and Exhaust Valve Construction and Operation.

machined out of the solid bar of steel until the thickness of the walls is only 1.5 millimeters—.05905 inch, or practically 1/16 inch. Each one has twenty-two fins which gradually taper down as the region of greatest pressure is departed from. In addition to carrying away heat, the fins assist in strengthening the walls of the cylinder. The barrel of the cylinder is slipped into the hole bored for it on the circumference of the crankcase and secured by a locking member in the nature of a stout compression ring, sprung onto a groove on the base of the cylinder within the crank chamber. On each lateral face of the crank chamber are seven holes, drilled right through the chamber parallel with the crankshaft. Each one of these holes receives a stout locking-pin of such a diameter that it presses against the split rings of two adjacent cylinders; in addition each cylinder is fitted with a key-way. This construction is not always followed, some of the early Gnome engines using the same system of cylinder retention as used on the latest "monosoupape" pattern.

Exhaust Valve Mounting.—The exhaust valve is mounted in the cylinder head, Fig. 376, its seating being screwed in by means of a special box spanner. On the fourteen-cylinder model the valve is operated directly by an overhead rocker arm with a gun metal rocker at its extremity coming in contact with the extremity of the valve stem. As in standard motor-car practice, the valve is opened under the lift of the vertical push rod, actuated by the cam. The distinctive feature is the use of a four-blade leaf spring with a forked end encircling the valve stems and pressing against a collar on its extremity. On the seven-cylinder model the movement is reversed, the valve being opened on the downward pull of the push rod, this lifting the outer extremity of the main rocker arm, which tips a secondary and smaller rocker arm in direct contact with the extremity of the valve stem. The springs are the same in each case. The two types are compared at A and B, Fig. 377.

Pistons Carry Inlet Valves.—The pistons, like the cylinders, are machined out of the solid bar of nickel steel, and have a portion of their wall cut away, so that the two adjacent ones will not come together at the extremity of their stroke. The head of the piston is slightly reduced in diameter and is provided with a groove into which is fitted a very light L-section brass split ring; back of this ring and carried within the groove is sprung a light steel compression ring, serving to keep the brass ring in expansion. As already mentioned, the intake valves are automatic, and are mounted in the head of the piston as outlined at Fig. 377 C. The valve seating is in halves, the lower portion being made to receive the wristpin and connecting rod, and the upper portion, carrying the valve, being screwed into it. The spring is composed of four flat blades, with the hollowed stem of the automatic valve passing through their center and their two extremities attached to small levers calculated to give balance against centrifugal force. The springs are naturally within the piston, and are lubricated by splash from the crank chamber. They are of a delicate construction, for it is necessary that they shall be accurately balanced so as to have no tendency to fly open under the action of centrifugal force. The intake valve is withdrawn by the use of special tools through the cylinder head, the exhaust valve being first dismounted.

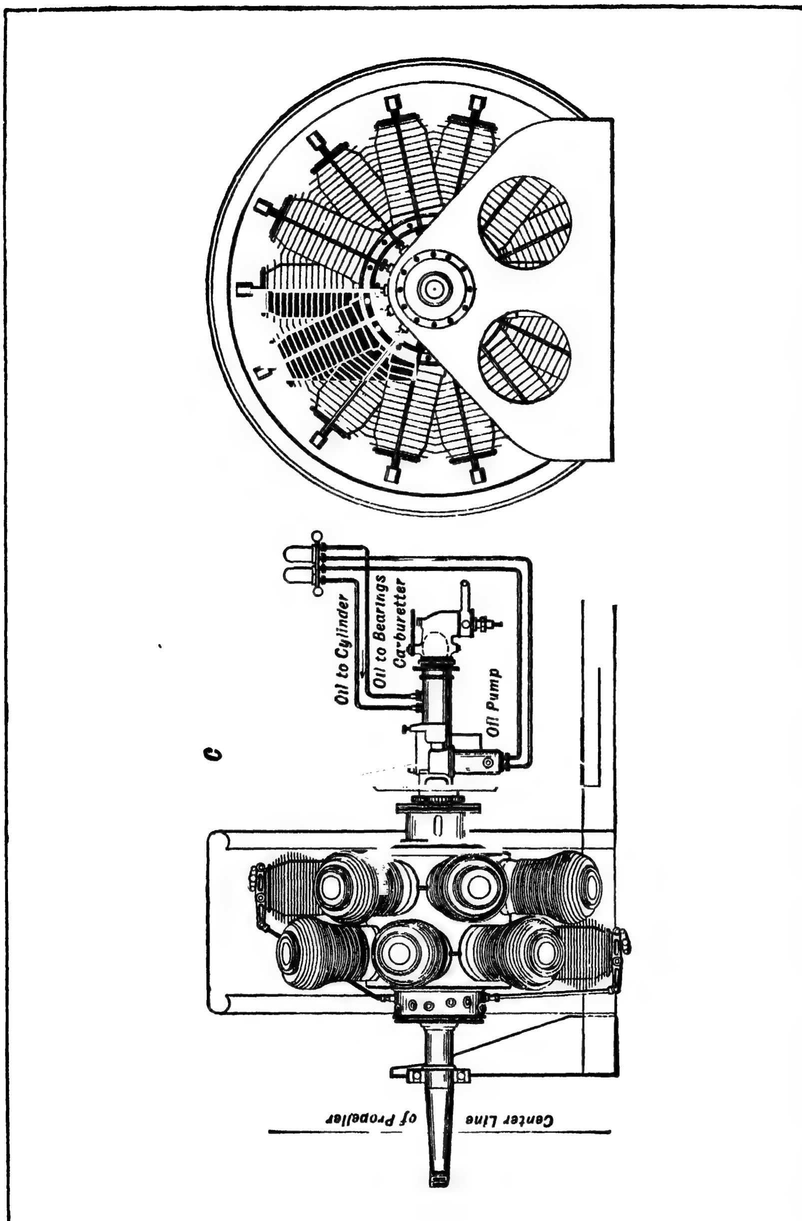


Fig. 378.—Installation Diagram Showing Method of Mounting an Early Fourteen-Cylinder, 100 Horsepower Gnome Aviation Engine.

Connecting Rod Arrangements.—The fourteen-cylinder motor shown at Fig. 378, has a two-throw crankshaft with the throws placed at 180 degrees, each one receiving seven connecting rods. The parts are the same as for the seven-cylinder motor, the larger one consisting of two groups placed side by side. For each group of seven-cylinders there is one main connecting rod, together with six auxiliary rods. The main connecting rod, which, like the others, is of H section, has machined with it two L-

section rings bored with six holes—51½ degrees apart to take the six other connecting rods. The cage of the main connecting rod carries two ball races, one on either side, fitting onto the crankpin and receiving the thrust of the seven connecting rods. The auxiliary connecting rods are secured in position in each case by a hollow steel pin passing through the two rings. It is evident that there is a slightly greater angularity for the six shorter

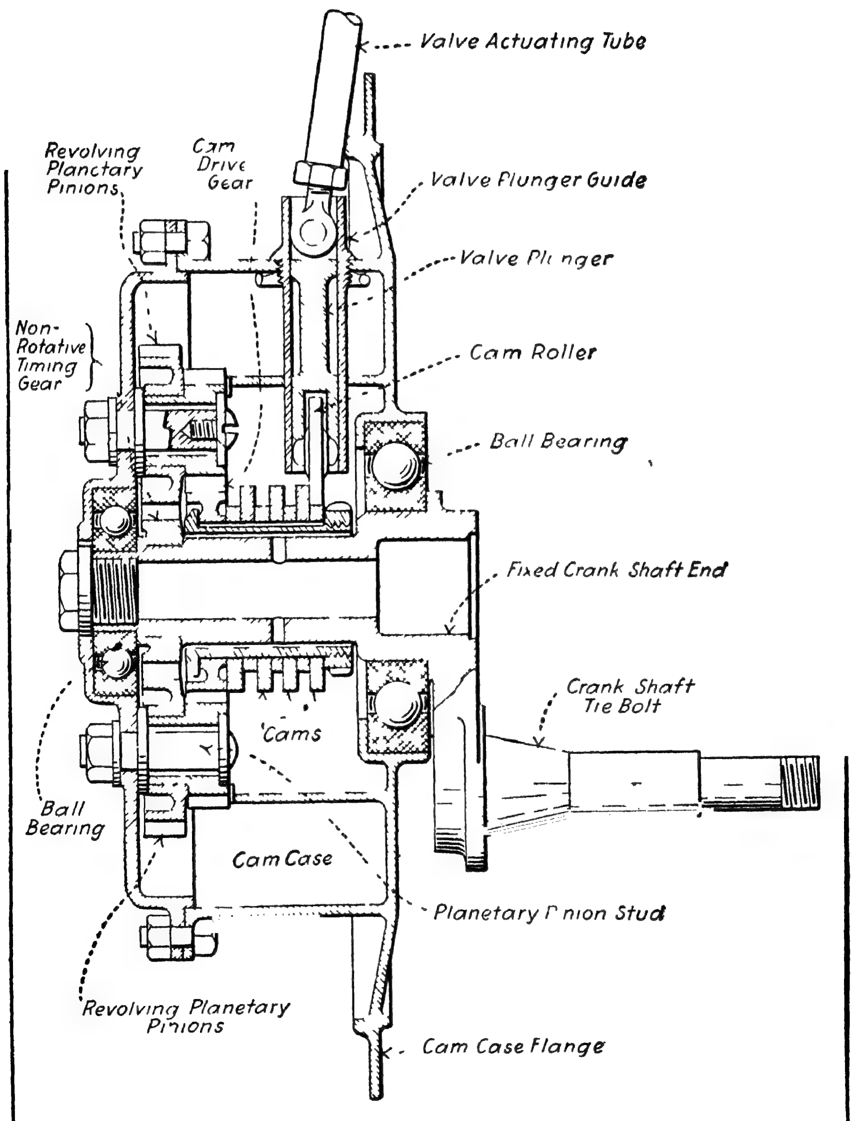


Fig. 379.—Cam and Cam Gearcase of the Gnome Seven-Cylinder Revolving Engine Showing Method of Driving Cams and Cam Follower Construction.

rods, known as auxiliary connecting rods, than for the longer main rods; this does not appear to have any influence on the running of the motor.

Exhaust Valve Operation.—Coming to the manner in which the earliest design exhaust valves are operated on the old style motor, this at first sight appears to be one of the most complicated parts of the motor, probably because it is one in which standard practice is most widely departed from. Within the cylindrical casing bolted to the rear face of the crankcase are seven, thin flat-faced steel rings forming female cams. Across a diameter of each ring is a pair of projecting rods fitting in brass guides and having their extremities terminating in a knuckle eye receiving the adjustable push rods operating the overhead rocker arms of the exhaust valve. The guides are not all in the same plane, the difference being equal to the thickness of the steel rings, the total thickness being practically two inches. Within the female cams is a group of seven male cams of the same total

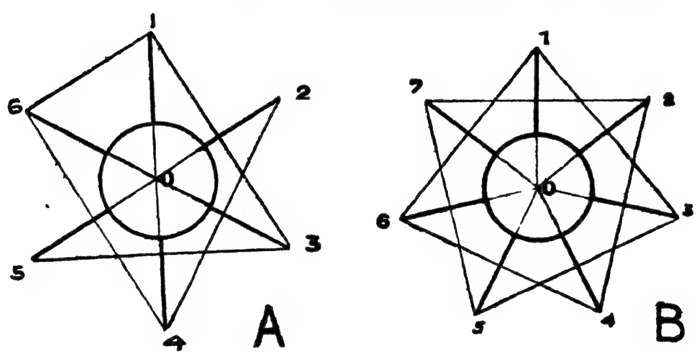


Fig. 380.—Diagram Showing Why an Odd Number of Cylinders is Best for Rotary or Static Radial Cylinder Motors with Single Crankpin.

thickness as the former and rotating within them. As the boss of the male cam comes into contact with the flattened portion of the ring forming the female cam, the arm is pushed outward and the exhaust valve opened through the medium of the push rod and overhead rocker. This construction was afterwards changed to seven male cams and simple valve operating plunger and roller cam followers as shown at Fig. 379.

On the face of the crankcase of the fourteen-cylinder motor opposite to the valve mechanism is a bolted-on end plate, carrying a pinion for driving the two magnetos and the two oil pumps, and having bolted to it the distributor for the high-tension current. Each group of seven-cylinders has its own magneto and lubricating pump. The two magnetos and the two pumps are mounted on the fixed platform carrying the stationary crankshaft, being driven by the pinion on the revolving crank chamber. The magnetos are geared up in the proportion of four to seven. Mounted on the end plate of the driving pinion are the two high-tension distributor plates, each one with seven brass segments let into it and connection made to the plugs by means of plain brass wire. The wire passes through a hole in the plug and is then wrapped round itself, giving a loose connection.

Why Odd Numbers of Cylinders are Used.—A good many people doubtless wonder why rotary and static radial engines are usually provided with an odd number of cylinders in preference to an even number. It is a matter of even torque, as can easily be understood from the accompanying diagram. Fig. 380 A represents a six-cylinder, single crank rotary engine, the radial lines indicating the cylinders. It is possible to fire the charges in two ways, firstly, in rotation, 1, 2, 3, 4, 5, 6, thus having six impulses in one revolution and none in the next; or alternately, 1, 3, 5, 2, 4, 6, in which case the engine will have turned through an equal number of degrees between impulses 1 and 3, and 3 and 5, but a greater number between 5 and 2, even again between 2 and 4, 4 and 6, and a less number between

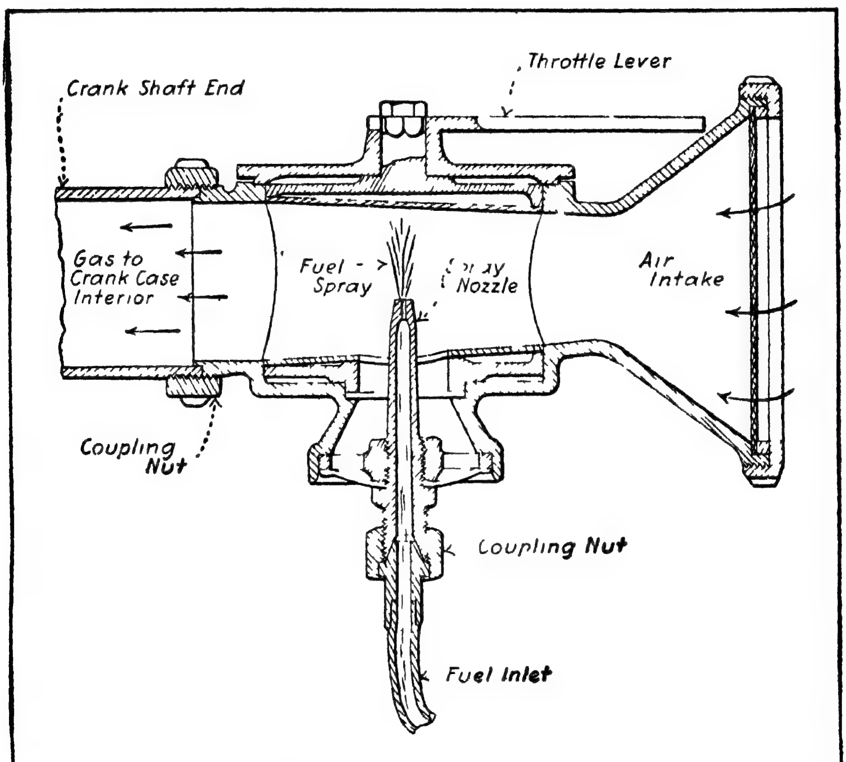


Fig. 381.—Diagram Showing Construction of the Simple Carburetor Used on Early Gnome Engines, Which was Attached to the Fixed Crankshaft End. Fuel Supply Was by Engine Driven Pump.

6 and 1, as will be clearly seen on reference to the diagram. Turning to Fig. 380 B, which represents a seven-cylinder engine. If the cylinders fire alternately it is obvious that the engine turns through an equal number of degrees between each impulse, thus, 1, 3, 5, 7, 2, 4, 6, 1, 3, etc. Thus supposing the engine to be revolving, the explosion takes place as each alternate cylinder passes, for instance, the point 1 on the diagram, and the ignition is actually operated in this way by a single contact. Six radial

cylinders may be used but a two throw crankshaft is necessary, each crankpin serving three cylinders.

Gnome Carburetion and Lubrication.—The crankshaft of the Gnome, as already explained, is fixed and hollow. For the seven- and nine-cylinder motors it has a single throw, and for the fourteen- and eighteen-cylinder models has two throws at 180 degrees. It is of the built-up type, this being necessary on account of the distinctive mounting of the connecting rods. The carburetor shown at Fig. 381 is mounted at one end of the stationary crankshaft, and the mixture is drawn in through a valve in the piston as already explained. There is neither float chamber nor jet. In many of the

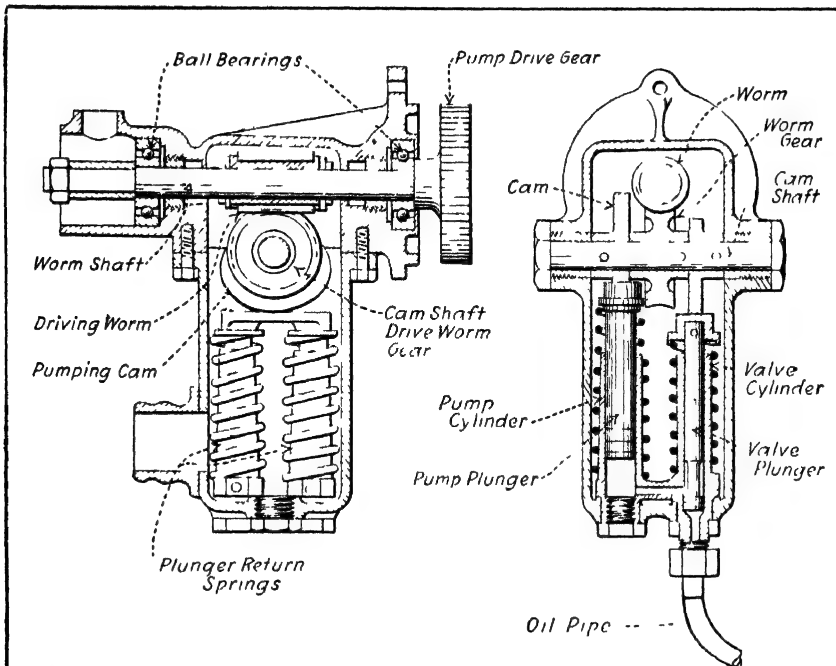


Fig. 382.—Sectional Views Showing Internal Construction of the Gnome Oil Pump.

tests made at the factory it is said the motor will run with the extremity of the gasoline pipe pushed into the hollow crankshaft, speed being regulated entirely by increasing or decreasing the flow through the shut-off valve in the base of the tank. Even under these conditions the motor has been throttled down to run at 350 revolutions without misfiring. Its normal speed is 1,000 to 1,200 revolutions a minute. Castor oil is used for lubricating the engine, the oil being injected into the hollow crankshaft through slight-feed fittings by a mechanically operated pump which is clearly shown in sectional diagrams at Fig. 382.

The Gnome is a considerable consumer of lubricant, the makers' estimate being seven pints an hour for the 100 horsepower motor; but in practice this is largely exceeded. The gasoline consumption is given as 300 to 350 grams per horsepower. The total weight of the fourteen-cylinder motor is

220 pounds without fuel or lubricating oil. Its full power is developed at 1,200 revolutions, and at this speed about nine horsepower is lost in overcoming air resistance to cylinder rotation.

Rotary Motor Disadvantages.—While the Gnome engine has many advantages, on the other hand, the head resistance offered by a motor of this type is considerable; there is a large waste of lubricating oil due to the centrifugal force which tends to throw the oil away from the cylinders; the gyroscopic effect of the rotary motor is detrimental to the best working of the airplane, and moreover, it requires about seven per cent of the total power developed by the motor to drive the revolving cylinders around the shaft. Of necessity, the compression of this type of motor is rather

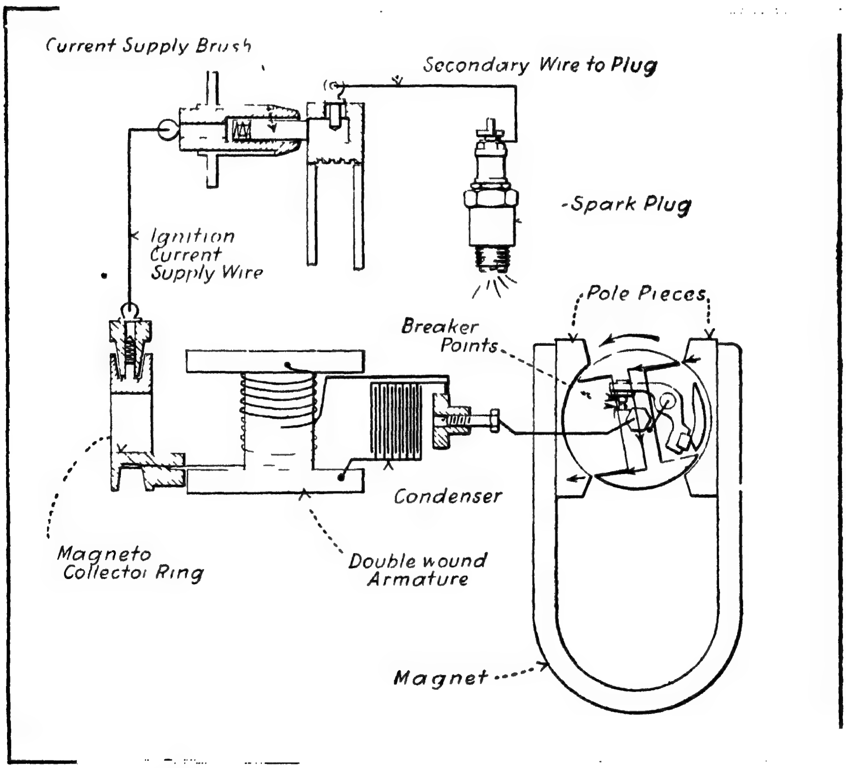


Fig. 383.—Simplified Diagram Showing the Gnome Motor Magneto Ignition System.

low, and an additional disadvantage manifests itself in the fact that there is as yet no satisfactory way of muffling the rotary type of motor.

Gnome "Monosoupape" Type.—The latest type of Gnome engine is known as the "monosoupape" type because but one valve is used in the cylinder head, the inlet valve in the piston being dispensed with on account of the trouble caused by that member on earlier engines. The construction of this latest type follows the lines established in the earlier designs to some extent and it differs only in the method of charging. The very rich mixture of gas and air is forced into the crankcase through the jet inside the crankshaft, and enters the cylinder when the piston is at its lowest position,

through the half-round openings in the guiding flange and the small holes or ports machined in the cylinder and clearly shown at Fig. 385. The returning piston covers the port, and the gas is compressed and fired in the usual way. The exhaust is through a large single valve in the cylinder head, which gives rise to the name "monosoupape," or single-valve motor, and this valve also remains open a portion of the intake stroke to admit air into the cylinder and dilute the rich gas forced in from the crankcase interior.

Aviators who have used the early form of Gnome say that the inlet

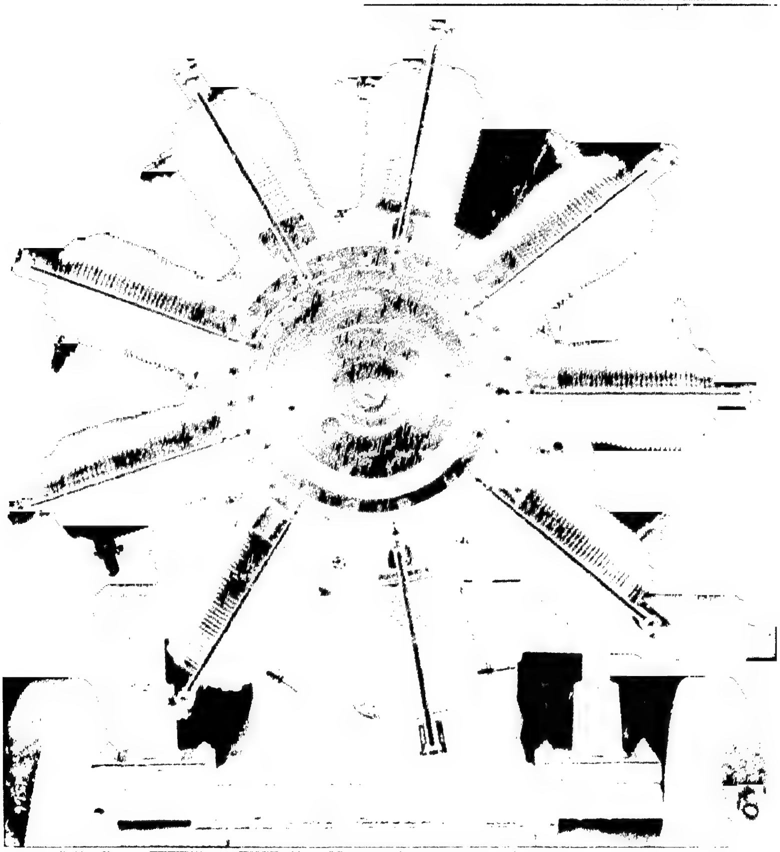


Fig. 384.—The War-Time General Vehicle Gnome "Monosoupape" Nine-Cylinder Rotary Engine, Mounted on Torque Stand for Testing Purposes.

valve in the piston type was prone to catch on fire if any valve defect materialized, but the "monosoupape" pattern is said to be nearly free of this danger. The bore of the 100 horsepower nine-cylinder engine is 110 millimeters, the piston stroke 150 millimeters. Extremely careful machine work and fitting is necessary. In many parts, tolerances of less than .0004 of an inch are all that are allowed. This is about one-sixth the thickness of the average human hair, and in other parts the size must be absolutely standard, no appreciable variation being allowable. The manufacture of this engine established new mechanical standards of engine production in

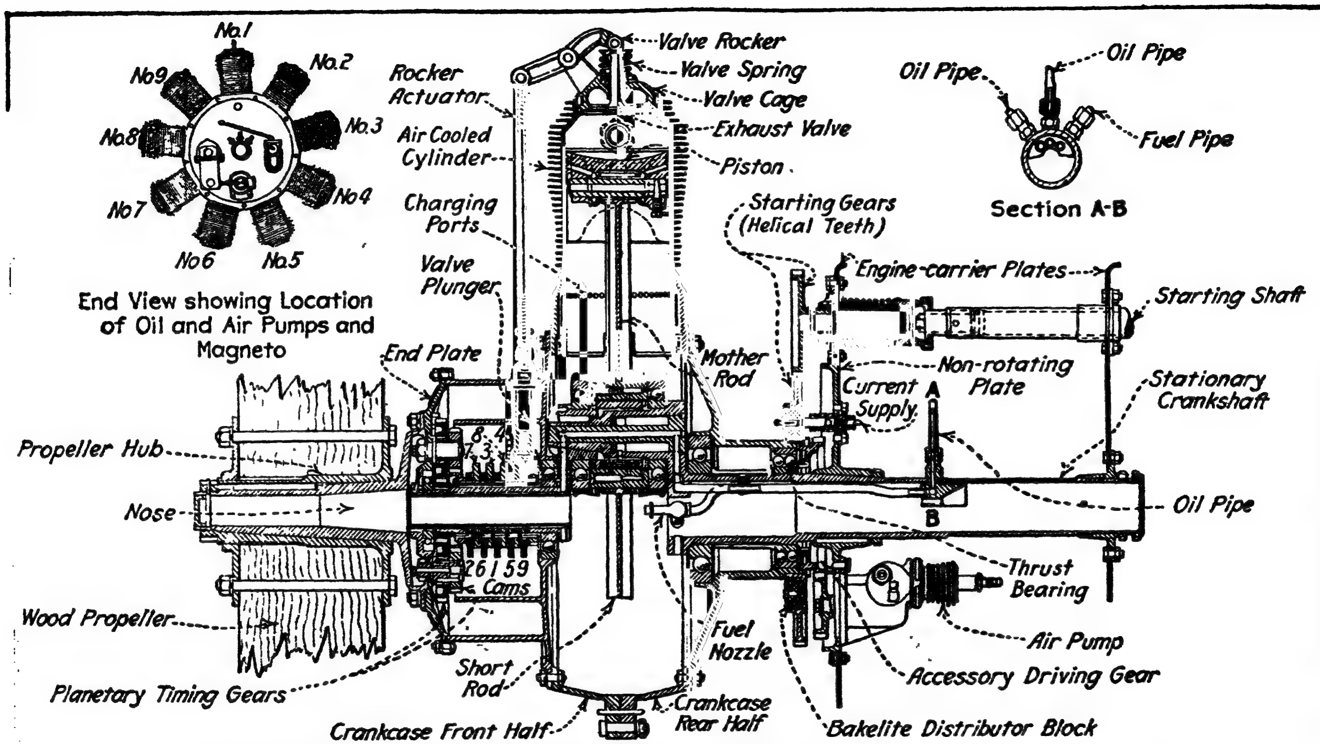


Fig. 385.—Longitudinal Sectional View Showing Construction of the General Vehicle Company "Monosoupape" Engine, a War-Time Type that is Now Obsolete.

this country. Much machine work was needed in producing the finished components from the bar and forging.

Details of Cylinder Construction.—The cylinders, for example, are machined from six inch solid steel bars, which are sawed into blanks eleven inches in length and weighing about 97 pounds. The first operation is to drill a $2\frac{1}{16}$ inch hole through the center of the block. A heavy-duty drilling machine performs this work, then the block goes to the lathe for further operations. Fig. 386 shows six stages of the progress of a cylinder, a few of the intermediate steps being omitted. These give, however, a good

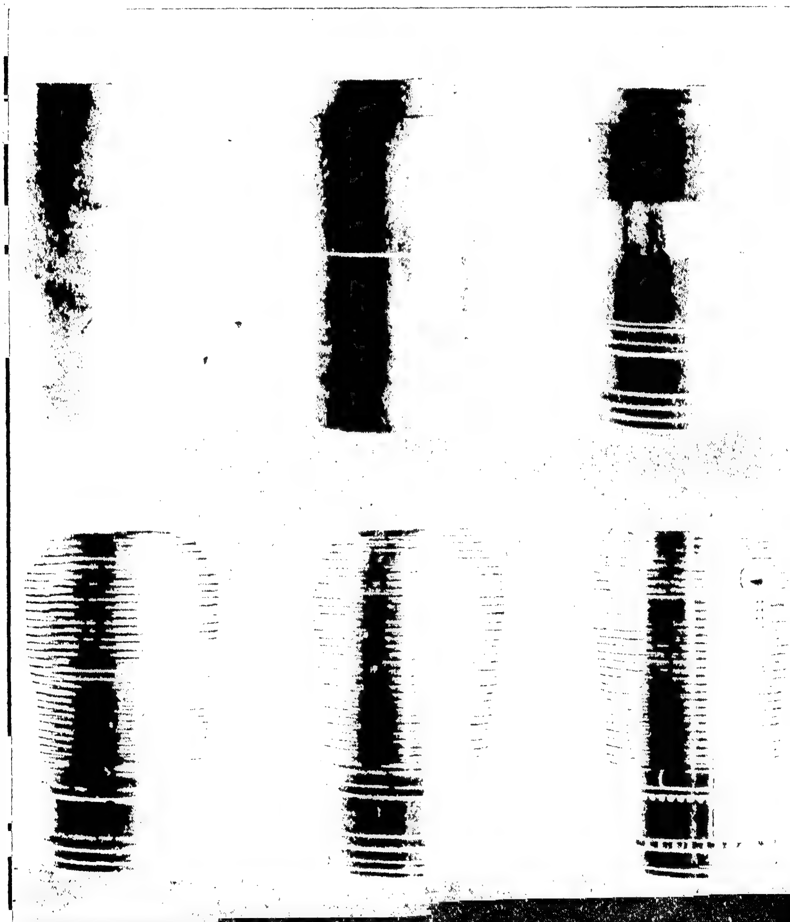


Fig. 386.—How a Gnome Engine Cylinder was Produced from a Solid Cylindrical Block of Chrome-Nickel Steel Weighing 97 Pounds, by Machining to the Finished Cylinder Weighing but Five and One-Half Pounds. This Method of Construction is Very Costly.

idea of the work done. The turning of the gills, or cooling flanges, is a difficult proposition, owing to the depth of the cut and the thin metal that forms the gills This operation requires the utmost care of tools and

the use of a good lubricant to prevent the metal from tearing as the tools approach their full depth. These gills are only 0.6 of a millimeter, or 0.0237 of an inch, thick at the top, tapering to a thickness of 1.4 millimeters (0.0553 of an inch) at the base, and are sixteen millimeters (0.632 of an inch) deep. When the machine work is completed the cylinder weighs but 5½ pounds.

Gnome "Monosoupape" Fuel System.—Gasoline is fed to the engine by means of air pressure at five pounds per square inch, which is produced by the air pump on the engine clearly shown at Fig. 385. A pressure gauge convenient to the operator indicates this pressure, and a valve enables the operator to control it. No carburetor is used. The gasoline flows from the tank through a shut-off valve near the operator and through a tube leading through the hollow crankshaft to a spray nozzle located in the

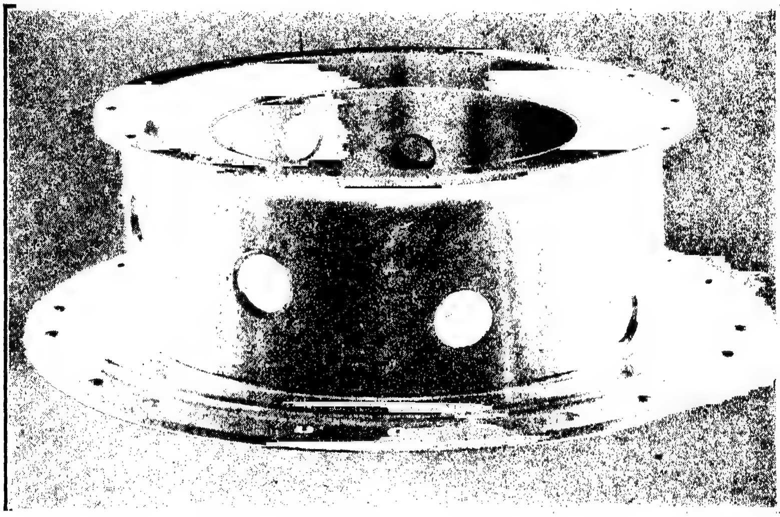


Fig. 387.—The Gnome Engine Alloy Steel Cam Gearcase was a Fine Example of Accurate but Expensive Machine Work, as it was Produced from a Heavy Forging. Corresponding Parts of Modern Engines Would be Made of Duraluminum.

crankcase. There is no throttle valve, and as each cylinder always receives the same amount of air as long as the atmospheric pressure is the same, the output cannot be varied by reducing the fuel supply, except within narrow limits. A fuel capacity of 65 gallons is provided. The fuel consumption is at the rate of twelve U. S. gallons per hour.

"Monosoupape" Ignition.—The high-tension magnetos, with double cam or two break per revolution interrupter, is located on the thrust plate in an inverted position, and is driven at such a speed as to produce nine sparks for every two revolutions; that is, at 2¼ times engine speed. A Splitdorf magneto is fitted. There is no distributor on the magneto. The high-tension collector brush of the magneto is connected to a distributor brush holder carried in the bearer plate of the engine. The brush in this brush holder is pressed against a distributor ring of insulating material moulded in position in the web of a gear wheel keyed to the thrust plate,

which gear serves also for starting the engine by hand. Moulded in this ring of insulating material are nine brass contact sectors, connecting with contact screws at the back side of the gear, from which bare wires connect to the sparkplugs. The distributor revolves at engine speed, instead of at

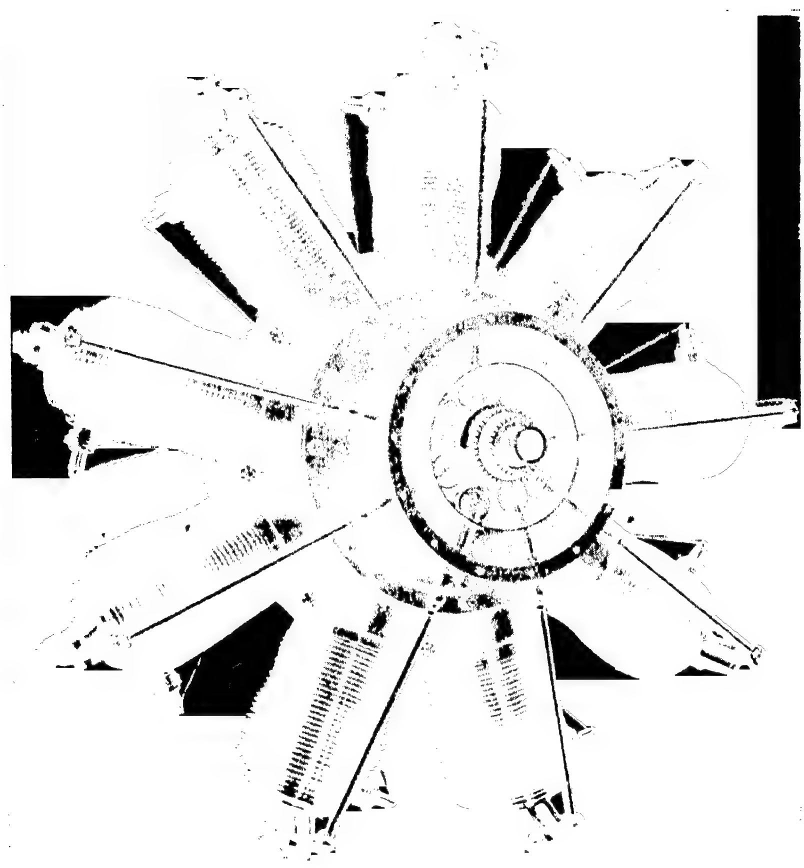


Fig. 388.—View of the Gnome "Monosoupape" Engine with Camcase Cover Removed to Show Cams and Valve Operating Plunger with Roller Cam Follower.

half engine speed as on ordinary engines, and the distributor brush is brought into electrical connection with each sparkplug every time the piston in the cylinder in which this sparkplug is located approaches the outer dead center. However, on the exhaust stroke no spark is being generated in the magneto, hence none is produced at the sparkplug.

Ordinarily the engine is started by turning on the propeller, but for emergency purposes as in seaplanes or for a quick "get away" if landing inadvertently in enemy territory, a hand starting crank was provided. This is supported in bearings secured to the pressed steel carriers of the engine and is provided with a universal joint between the two supports so as to prevent binding of the crank in the bearings due to possible distortion of

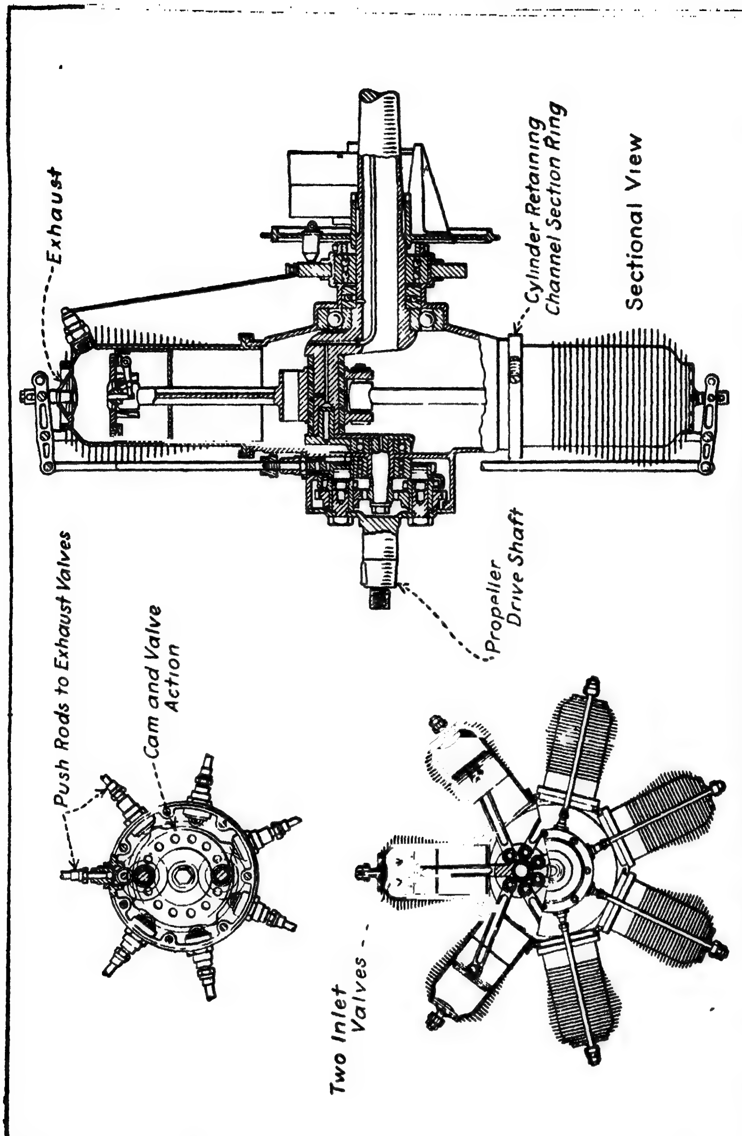


Fig. 389.—The 50 Horsepower Rotary Bayerishen Motoren Gesellschaft Engine Was a German Adaptation of the Early Gnome Design.

the supports. The gear on this starting crank and the one on the thrust plate with which it meshes are cut with helical teeth of such hand that the starting pinion is thrown out of mesh as soon as the engine picks up its cycle. A coiled spring surrounds part of the shaft of the starting crank and holds it out of gear when not in use.

"Monosoupape" Lubrication.—Lubricating oil is carried in a tank of 25 gallon capacity, and if this tank has to be placed in a low position it is connected with the air-pressure line, so that the suction of the oil pump is not depended upon to get the oil to the pump. From the bottom of the oil tank a pipe leads to the pump inlet. There are two outlets from the

pump, each entering the hollow crankshaft, and there is a branch from each outlet pipe to a circulation indicator convenient to the operator. One of the oil leads feeds to the housings in the thrust plate containing the two rear ball bearings, and the other lead feeds through the crankpin to the cams, as already explained.

Owing to the effect of centrifugal force and the fact that the oil is not used over again, the oil consumption of a revolving cylinder engine is considerably higher than that of a stationary cylinder engine. Fuel consumption is also somewhat higher, and for this reason the revolving cylinder engine is not so well suited for types of airplanes designed for long

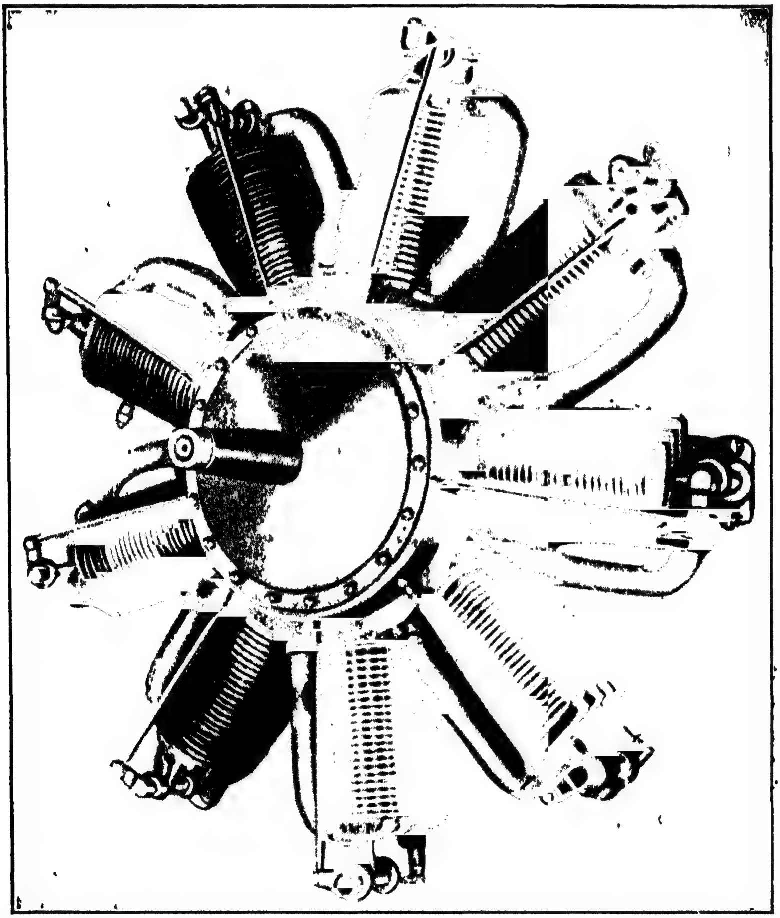


Fig. 390.—View Showing Construction of the Nine-Cylinder Revolving LeRhone Type Aviation Engine.

trips, as the increased weight of supplies required for such trips, as compared with stationary cylinder type motors, more than offsets the high weight efficiency of the engine itself. But for short trips, and especially where high speed is required, as in single-seated scout and battle planes or "avions de chasse," as the French say, the revolving cylinder engine had

the advantage. The oil consumption of the Gnome engine is as high as 2.4 gallons per hour. Castor oil is used for lubrication because it is not cut by the gasoline mist present in the engine interior as an oil of mineral derivation would be.

German "Gnome" Type Engine.—A German adaptation of the Gnome design is shown at Fig. 389. This is known as the Bayerischen Motoren Gesellshaft engine and the type shown is an early design rated at 50 horsepower. The bore is 110 millimeters, the stroke is 120 millimeters, and it is designed to run at a speed of 1,200 r.p.m. It is somewhat similar in design to the early Gnome "valve-in-piston" design except that two valves are carried in the piston top instead of one. The valve operating arrangement is different also, as a single four point cam is used to operate the seven exhaust valves. It is driven by epicyclic gearing, the cam being driven by an internal gear machined integrally with it, the cam being turned at ⅞ times the engine speed. Another feature is the method of holding the cylinders on the crankcase. The cylinder is provided with a flange that registers with a corresponding member of the same diameter on the crankcase. A U section, split clamping ring is bolted in place as shown, this holding both flanges firmly together and keeping the cylinder firmly seated against the crankcase flange. The "monosoupape" type has also been copied and has received some application in Germany, but the most successful early German airplanes were powered with six-cylinder vertical engines such as the Benz and Mercedes.

The Le Rhone Rotary Motor.—The Le Rhone motor is a radial revolving cylinder engine that has many of the principles which were incorporated in the Gnome but which were considered to be an improvement by many foreign aviators. Instead of having but one valve in the cylinder head, as the latest type "monosoupape" Gnome has, the Le Rhone has two valves, one for intake and one for exhaust in each cylinder. By an ingenious rocker arm and tappet rod arrangement it is possible to operate both valves with a single push rod. Inlet pipes communicate with the crankcase at one end and direct the fresh gas to the inlet valve cage at the other. Another peculiarity in the design is the method of holding the cylinders in place. Instead of having a vertically divided crankcase as the Gnome engine has and clamping both halves of the case around the cylinders, the crankcase of the Le Rhone engine is in the form of a cylinder having nine bosses provided with threaded openings into which the cylinders are screwed. A thread is provided at the base of each cylinder and when the cylinder has been screwed down the proper amount it is prevented from further rotation about its own axis by a substantial lock nut which screws down against the threaded boss on the crankcase. The external appearance of the Le Rhone type motor is clearly shown at Fig. 390, while the general features of construction are clearly outlined in the sectional views given at Figs. 391 and 392.

Le Rhone Connecting Rod Arrangement.—The two main peculiarities of this motor are the method of valve actuation by two large cams and the distinctive crankshaft and connecting rod big end construction. The connecting rods are provided with "feet" or shoes on the end which fit into grooves lined with bearing metal which are machined into crank discs

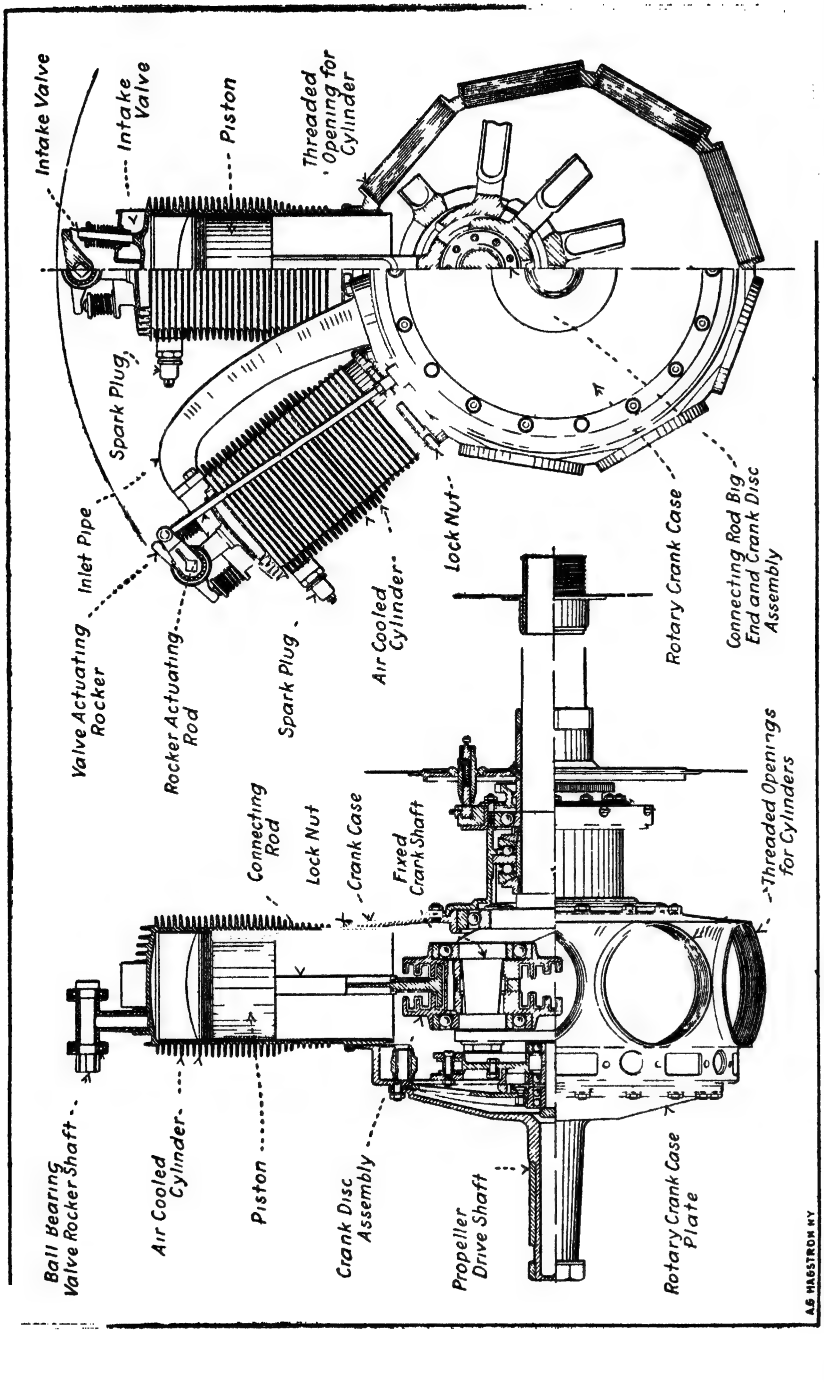


Fig. 391.—Part Sectional Views of the LeRhone Rotary Cylinder Engine, Showing Method of Cylinder Retention, Valve Operation, and Novel Crankdisc and Connecting Rod Big End Assembly.

revolving on ball bearings and which are held together so that the connecting rod big ends are sandwiched between them by clamping screws. This construction is a modification of that used on the Anzani six-cylinder radial engine. There are three grooves machined in each crank disc and three connecting rod big ends run in each pair of grooves. The details of this construction can be readily ascertained by reference to explanatory diagrams at Figs. 393 and 394 A. Three of the rods which work in the groove nearest the crankpin are provided with short shoes as shown at Fig. 394 B. The short shoes are used on the rods employed in cylinders number 1, 4, and 7. The set of connecting rods that work in the central

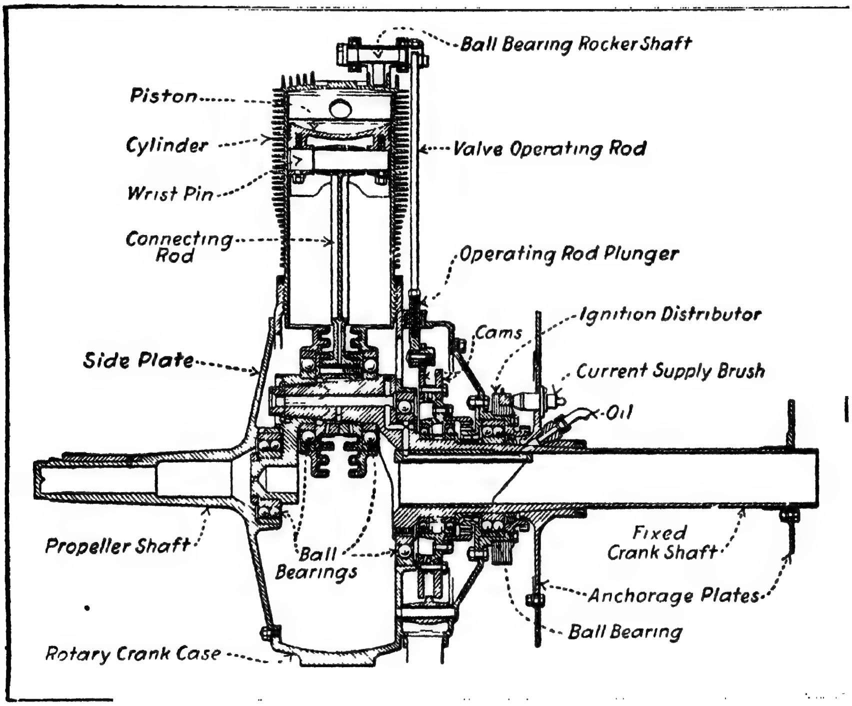


Fig. 392.—Side Sectional View of the LeRhone Fixed Crankshaft, Aviation Engine.

grooves are provided with medium-length shoes and actuate the pistons in cylinders numbers 3, 6, and 9. The three rods that work in the outside grooves have still longer shoes and are employed in cylinders numbers 2, 5, and 8.

Le Rhone Valve Actuation.—The peculiar profile of the inlet and exhaust cam plates are shown at C, Fig. 394, while the construction of the wristpin, wristpin bushing and piston are clearly outlined at the sectional view at E. The method of valve actuation is clearly outlined at Fig. 395, which shows an end section through the cam case and also a partial side elevation showing one of the valve operating levers which is fulcrumed at a central point and which has a roller at one end bearing on one cam while the roller or cam follower at the other end bears on the other cam. The

valve rocker arm actuating rod is, of course, operated by this simple lever and is attached to it in such a way that it can be pulled down to depress the inlet valve and pushed up to open the exhaust valve.

Le Rhone Carburetor.—A carburetor of peculiar construction is employed in the Le Rhone engine, this being a very simple type as outlined at Fig. 396. It is attached to the threaded end of the hollow crankshaft by a right and left coupling. The fuel is pumped to the spray nozzle, the opening in which is controlled by a fuel regulating needle having a long

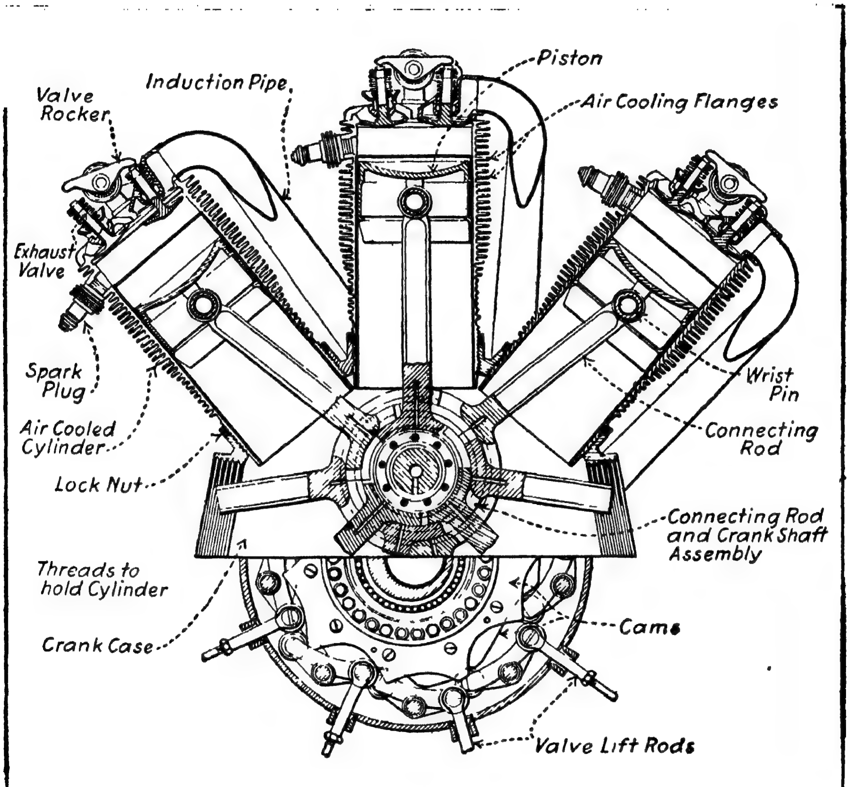


Fig. 393.—View Showing the LeRhone Valve Action and Connecting Rod Big End Arrangement.

taper which is lifted out of the jet opening when the air-regulating slide is moved. The amount of fuel supplied the carburetor is controlled by a special needle valve fitting which combines a filter screen and which is shown at B. In regulating the speed of the Le Rhone engine, there are two possible means of controlling the mixture, one by altering the position of the air-regulating slide, which also works the metering needle in the jet, and the other by controlling the amount of fuel supplied to the spray nozzle through the special fitting provided for that purpose.

Le Rhone Engine Action.—In considering the action of this engine one can refer to Fig. 397. The crank O, M, is fixed, while the cylinders can

turn about the crankshaft center O and the piston turns around the crank-pin M, because of the eccentricity of the centers of rotation the pistons will reciprocate in the cylinders This distance is at its maximum when the cylinder is above O and at a minimum when it is above M, and the difference between these two positions is equal to the stroke, which is twice the

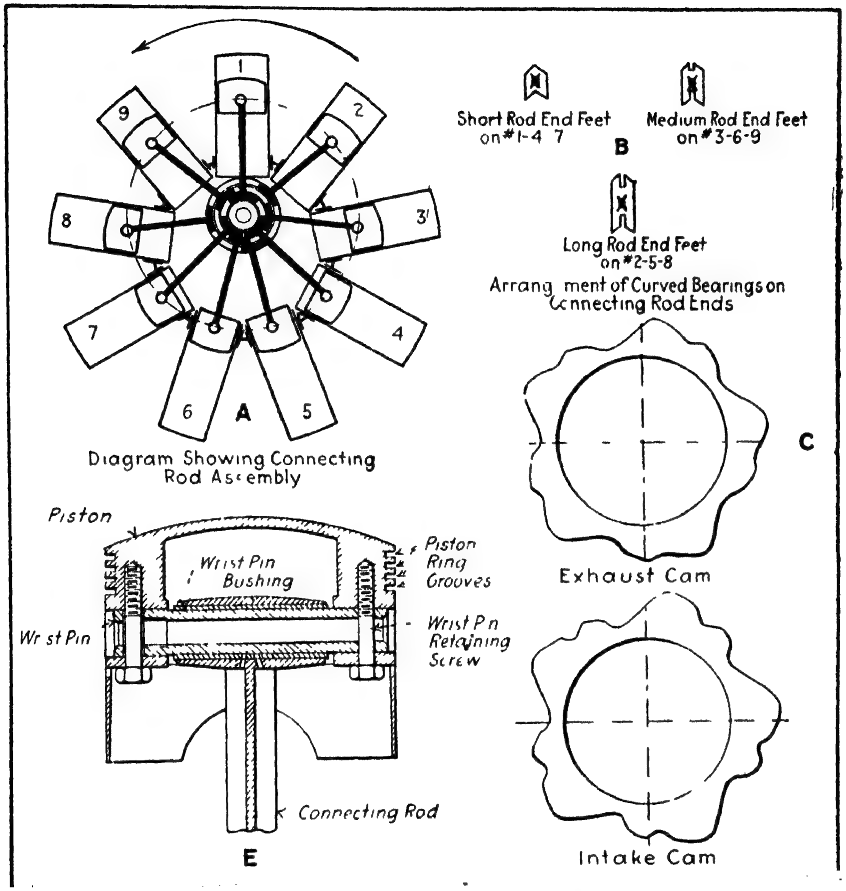


Fig. 394.—Diagrams Showing Construction of Important Components of the LeRhone Motor. Note Variation in Length of the Curved Bearings of Connecting Rod Big End, as Shown at B, and Peculiar Profile of Inlet and Exhaust Cams as Shown at C and D. Diagram at A Shows Connecting Rod Assembly and Crankrings. Sectional View at E Outlines Wristpin Retention and Piston Design.

distance of the crank-throw O, M. The explosion pressure resolves itself into the force F exerted along the line of the connecting rod A, M, and also into a force N, which tends to make the cylinders rotate around point O in the direction of the arrow. An odd number of cylinders acting on one crankpin is desirable to secure equally spaced explosions, as the basic action is the same as the Gnome engine.

The magneto is driven by a gear having 36 teeth attached to crankcase which meshes with sixteen-tooth pinion on armature. The magneto turns

at 2.25 times crankcase speed. Two cams, one for inlet, one for exhaust, are mounted on a carrying member and act on nine rocker arms which are capable of giving a push-and-pull motion to the valve-actuating rocker-operating rods. A gear driven by the crankcase meshes with a larger member having internal teeth carried by the cam carrier. Each cam has five profiles and is mounted in staggered relation to the other. These give the nine fulcrumed levers the proper motion to open the inlet and exhaust valves at the proper time. The cams are driven at $^{45}/_{50}$ or $^{9}/_{10}$ of the motor

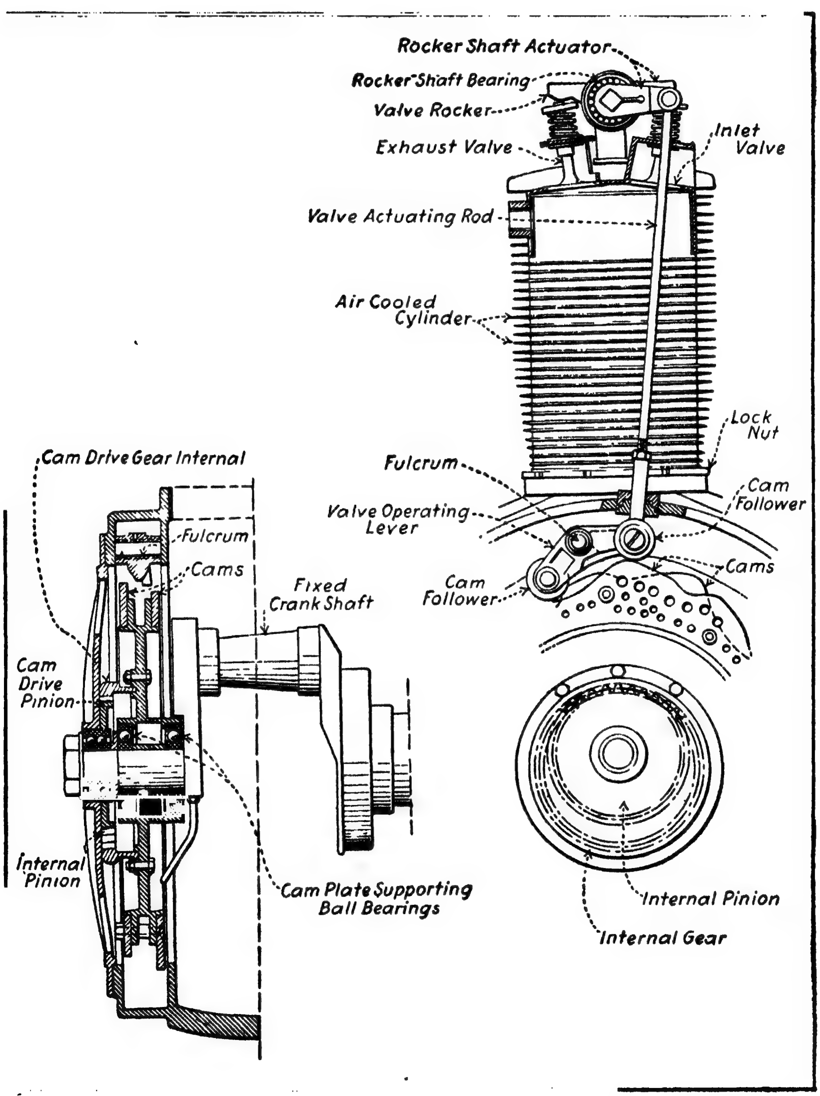


Fig. 395.—How the Cams of LeRhone Motor Can Operate Two Valves with a Single Push Rod, by Using Centrally Fulcrumed Double Cam Followers.

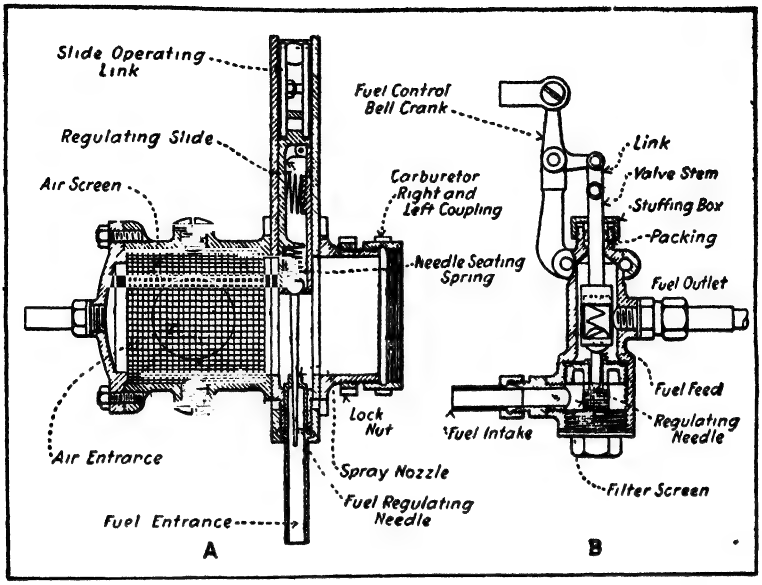


Fig. 396.—The LeRhone Carburetor at A, and Fuel Supply Regulation Devices at B.

speed. The cylinder dimensions and timing follows; the weight can be approximated by figuring three pounds per horsepower.

80 H. P.	105 M/M bore	4.20" bore
	140 M/M stroke	5.60" stroke
110 H. P.	112 M/M bore	4.48" bore
	170 M/M stroke	6.80" stroke

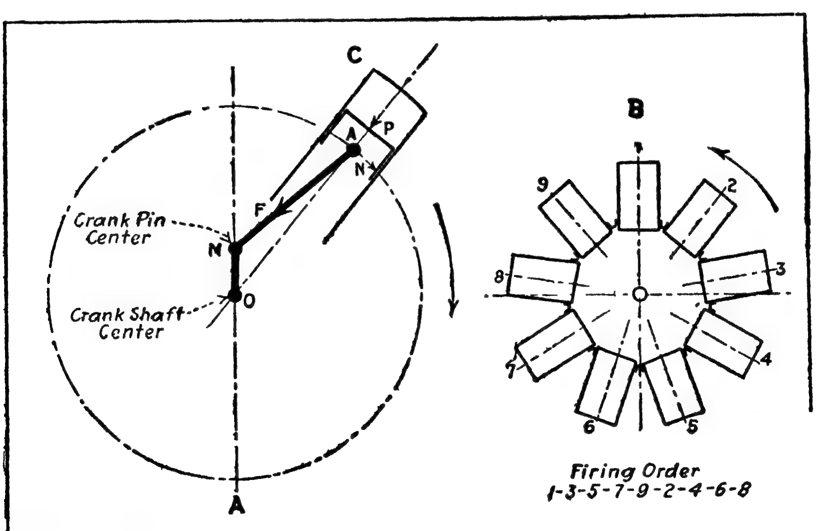


Fig. 397.—Diagram Showing LeRhone Motor Action and Cylinder Firing Order.

	110 H. P.	80 H. P.
Timing—Intake valve opening, lag	18°	18°
Intake valve closing, lag	35°	35°
Exhaust valve opening, lead	55°	45°
Exhaust valve closing, lag	5°	5°
Ignition time advance	26°	26°

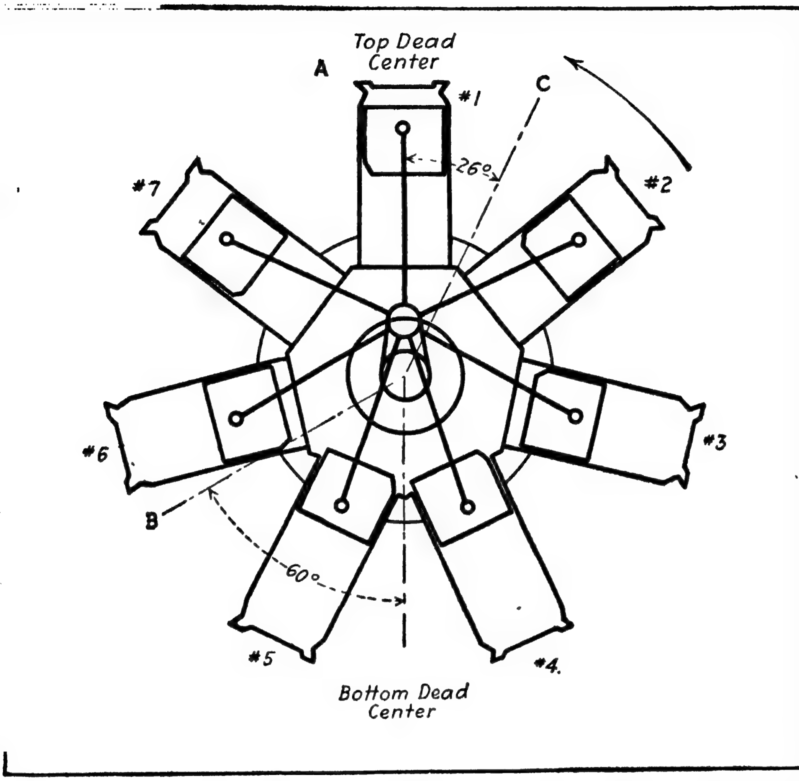


Fig. 398.—Diagram Showing Positions of Pistons in LeRhone Rotary Cylinder Motor when Piston in Cylinder No. 1 is at Top Dead Center.

The Clerget Engine.—Several air-cooled rotary types having seven, nine and eleven cylinders were built by Clerget, Blin and Cie of Paris and received considerable application during the World War on French and British pursuit planes and on training planes. As will be seen from the sectional drawing presented at Fig. 399 A this engine had some of the characteristics of the Gnome and LeRhone engines. The cylinder construction was similar to the LeRhone engine except that separate actuating means were provided for each valve. The master rod assembly was similar to that used in the Gnome engine. The cylinders were machined from solid billets of steel with integral cooling fins. These are held in place through the gripping action of the vertically divided crankcase clamping

a circumferential flange in an annulus made to receive it. The nose piece of the motor carries the propeller hub and the crankshaft is a separable type. At the anti-propeller end of the fixed crankshaft, which is of large diameter and hollow or tubular section, the fuel spray nozzle of the "bloc" tube carburetor is placed. Fuel is injected in by pump pressure, just as in the Gnome and LeRhone motors. Ball bearings are used liberally in the connecting rod assembly and for supporting the rotating crankcase from the fixed crankshaft. The pistons are fitted with three rings of the conventional type and two obdurator rings, the latter being carried in the

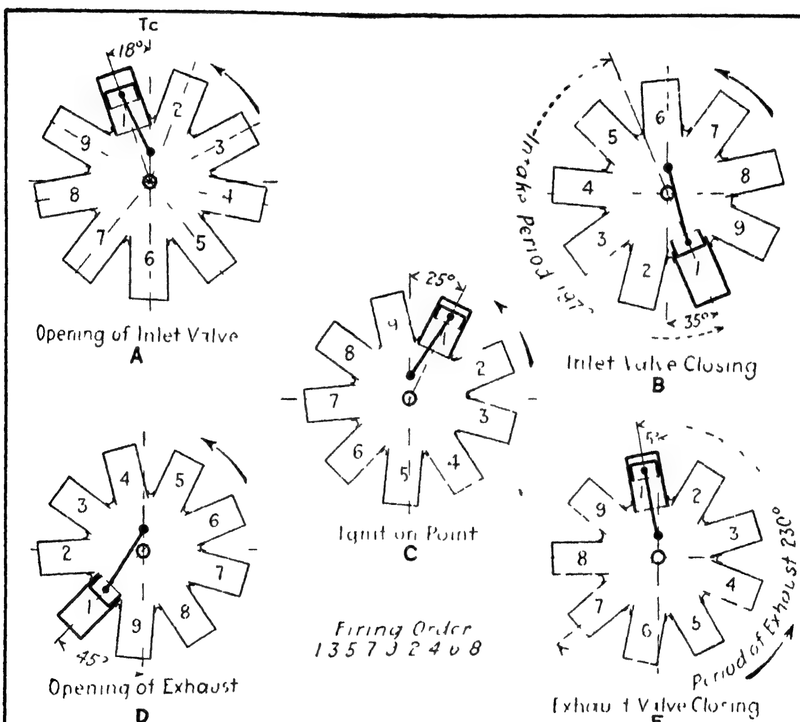


Fig. 399.—Diagrams Showing Valve Timing of LeRhone Aviation Engine.

same groove, one inside the other. The cam ring is rotated by an eccentric with teeth, a feature of design exclusively Clerget. The castor oil used for lubrication is circulated by plunger pumps. High-tension magnetos supply current to a fixed distributor brush which bears against segments on a back plate, each segment being connected to one of the sparkplugs in line with it. Bare copper wires conducted the current to the sparkplugs. The nine-cylinder Type 9Z had cylinders with a total displacement of 992 cubic inches. The bore was 4.72 inches, the stroke 6.3 inches. The engine developed 121 horsepower at 1,200 r.p.m. and 123 at 1,300 r.p.m. The compression ratio was 4.36 to 1. As in all rotary engines the fuel consumption was about .8 pound per horsepower-hour and the oil con-

sumption .15 pound per horsepower-hour. The dry weight was reported to be 367 pounds or 3.03 pounds per normal horsepower.

The Renault Air-Cooled Vee Engine.—Air-cooled stationary cylinder engines were rarely used in airplanes, but the Renault Frères of France for several years manufactured a complete series of such engines of the general design shown at Fig. 400, ranging from a low-powered one developed nineteen years ago and rated at 40 and 50 horsepower, to later

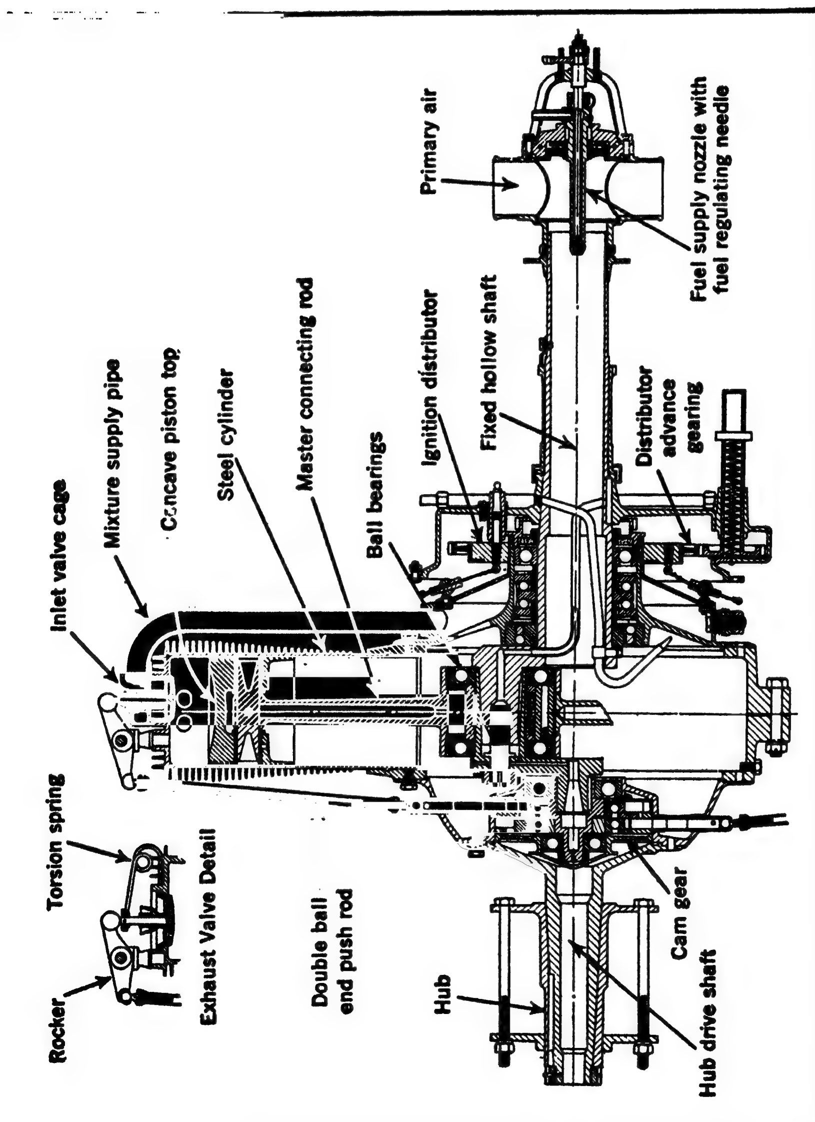


Fig. 399A.—Sectional View Showing Construction of Clerget 150 Horsepower Revolving Crankcase Motor.

eight-cylinder models rated at 70 horsepower and a twelve-cylinder, or twin six, rated at 90 horsepower. The cylinders are of cast iron and are furnished with numerous cooling ribs which are cast integrally. The cylinder heads are separate castings and are attached to the cylinder as in early motorcycle engine practice, and serve to hold the cylinder in place on the aluminum alloy crankcase by a cruciform yoke and four long hold-down bolts (Fig. 401). The pistons are of cast steel and utilize piston rings of cast iron. The valves are situated on the inner side of the cylinder head, the arrangment being unconventional in that the exhaust valves are placed above the inlet. The inlet valves seat in an extension of the combustion

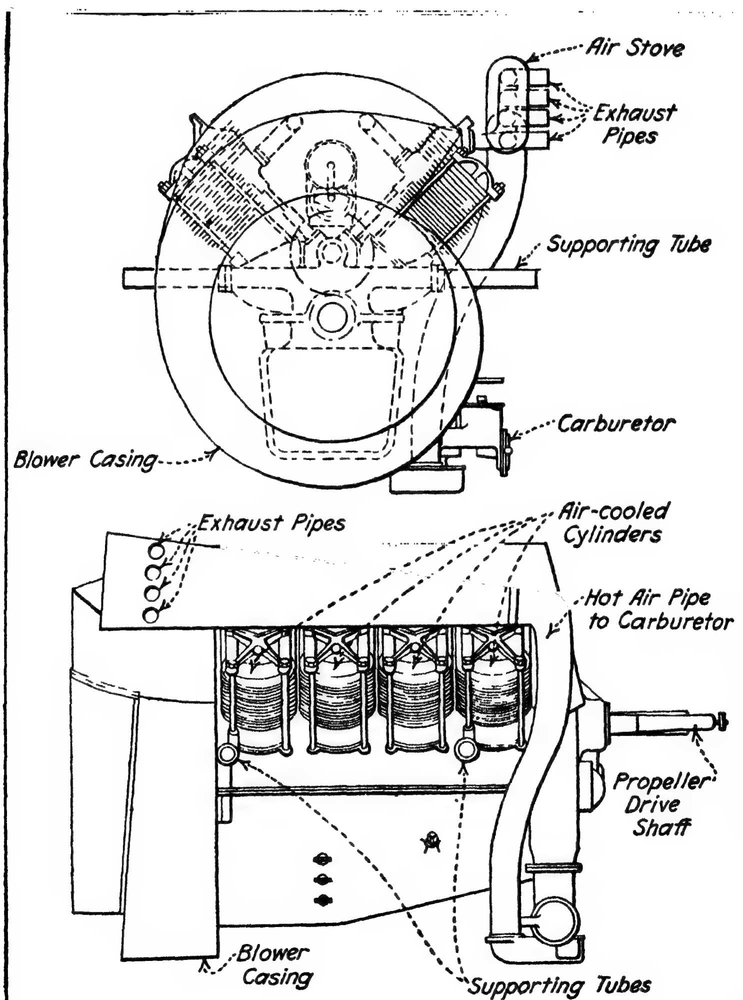


Fig. 400.—Diagram Showing How Cylinder Cooling is Effected in the Renault Eight-Cylinder "Vee" Engine.

head and are actuated by direct push rod and cam in the usual manner while an overhead gear in which rockers are operated by push rods is needed to actuate the exhaust valves. The valve action is clearly shown in Figs. 226 and 227. The air stream by which the cylinders are cooled is produced by a centrifugal or blower type fan of relatively large diameter which is mounted on the end of a crankshaft and the air blast is delivered from this blower into an enclosed space between the cylinder from which it escapes only after passing over the cooling fins. In spite of the fact that considerable prejudice existed against air-cooling fixed cylinder en-

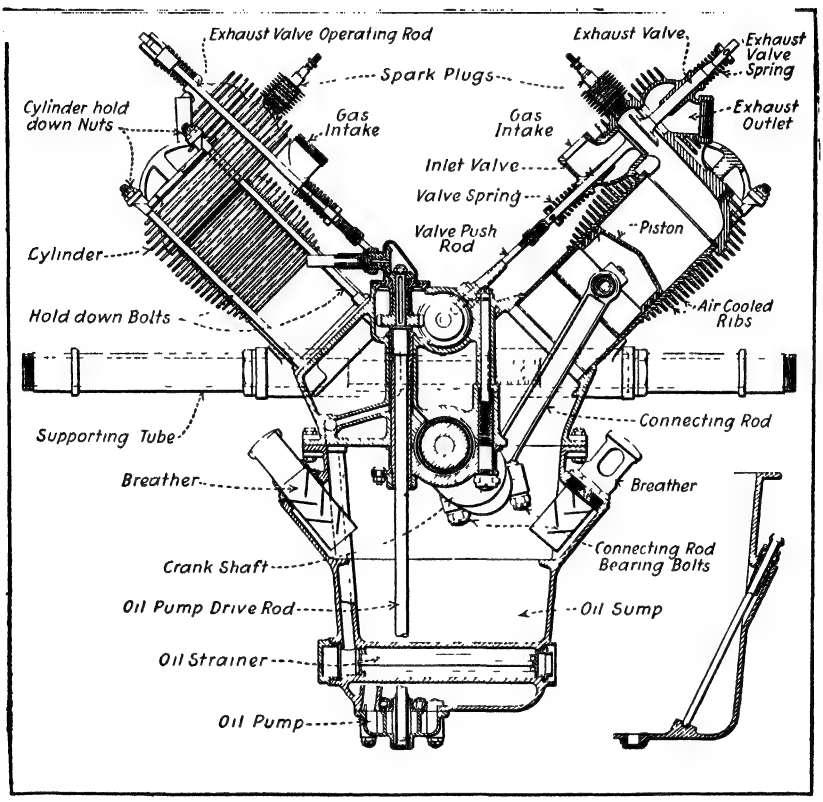


Fig. 401.—End Sectional View of Eight-Cylinder "Vee" Type Renault Air-Cooled Aviation Engine. This Type of Construction is Being Seriously Considered in a Refined and Simplified Form for Modern Airplanes.

gines, the Renault had given very good service in both England and France.

As will be seen by the sectional view at Fig. 402, the steel crankshaft is carried in a combination of plain bearings inside the crankcase and by ball bearings at the ends. Owing to air cooling, special precautions are taken with the lubrication system, though the lubrication is not forced or under high pressure. An oil pump of the gear-wheel type delivers oil from the sump at the bottom of the crankcase to a chamber above, from which

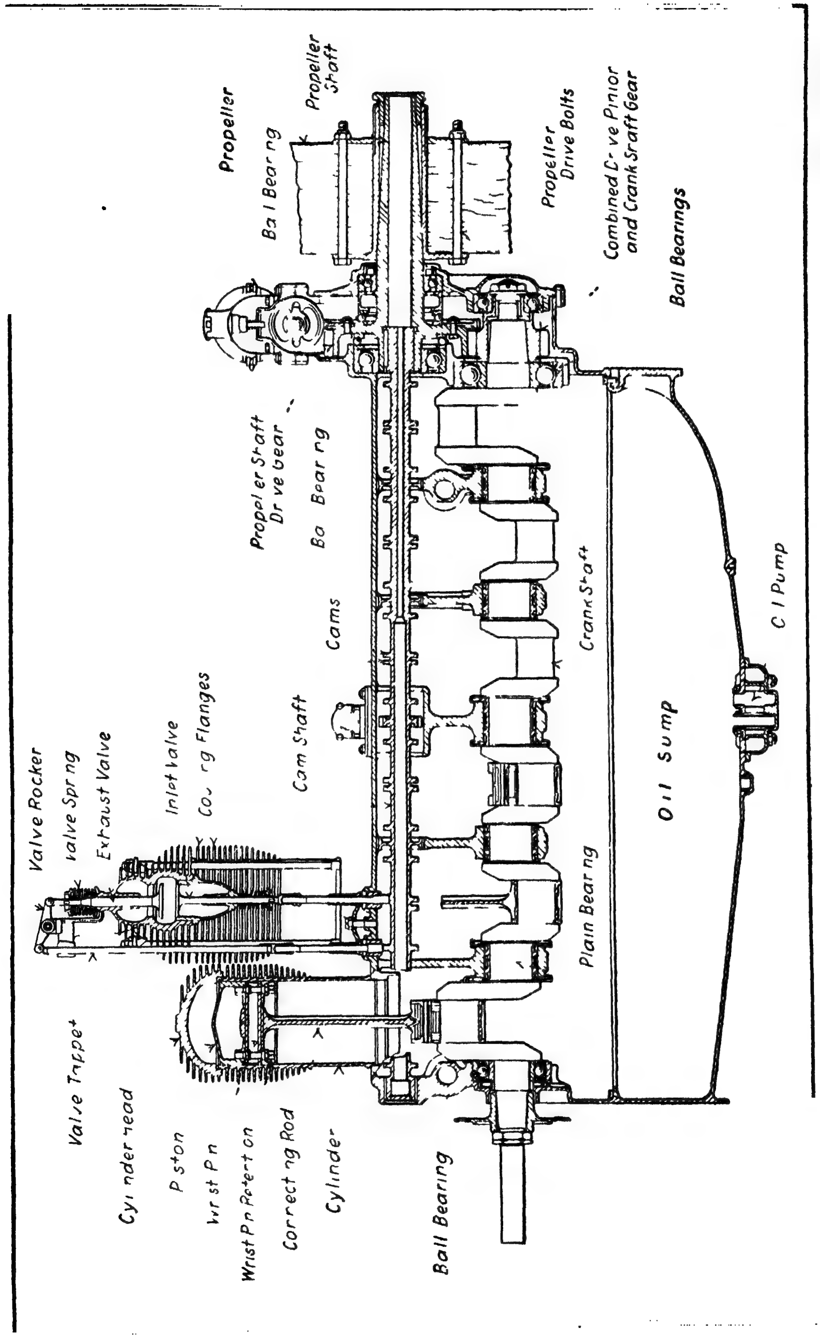


Fig. 402.—Side Sectional View of Renault Twelve-Cylinder Air-Cooled Aviation Engine Crankcase Showing the Use of Plain and Ball Bearings for Crankshaft Support.

the oil flows by gravity along suitable channels to the various main bearings. It flows from the bearings into hollow rings fastened to the crank-webs, and the oil thrown from the whirling connecting rod big ends bathes the internal parts in an oil mist.

In the eight-cylinder designs ignition is effected by a magneto giving four sparks per revolution and is accordingly driven at engine speed. In the twelve-cylinder machine two magnetos of the ordinary revolving armature or two-spark type, each supplying six cylinders, are fitted as outlined

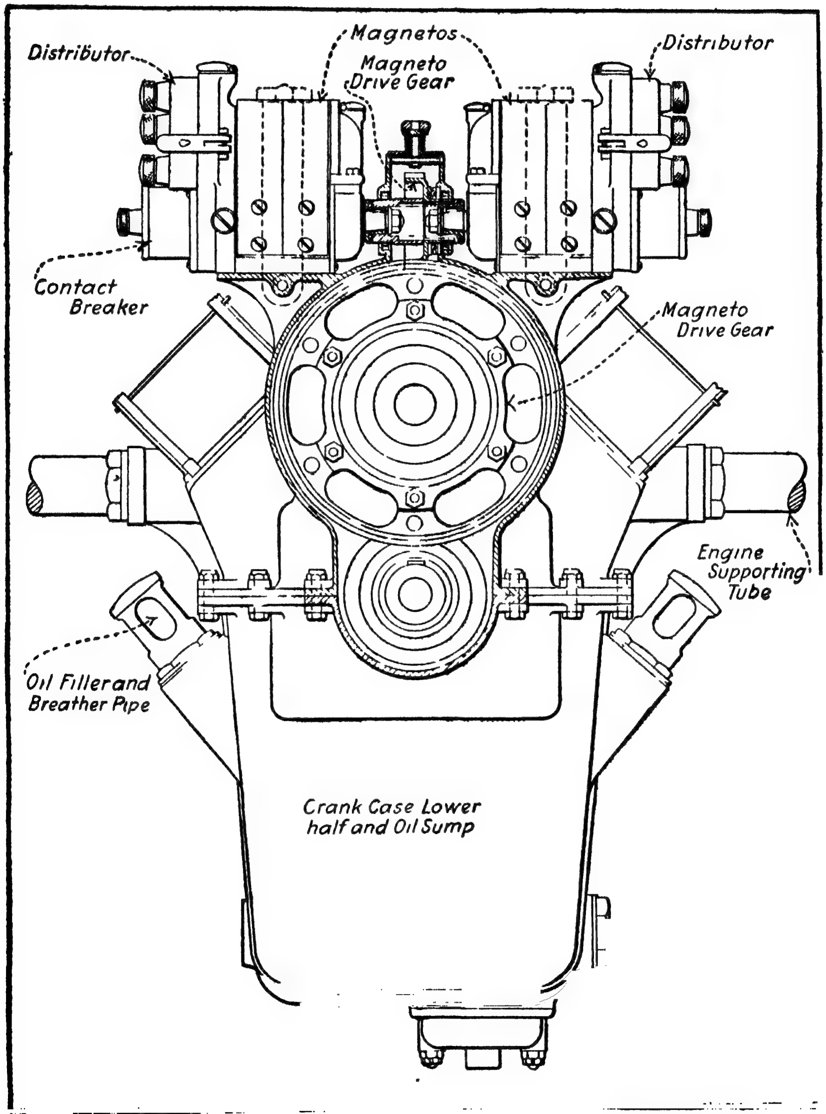


Fig. 403.—End View of Renault Twelve-Cylinder Engine Crankcase Showing Method of Magneto Drive and Magneto Mounting.

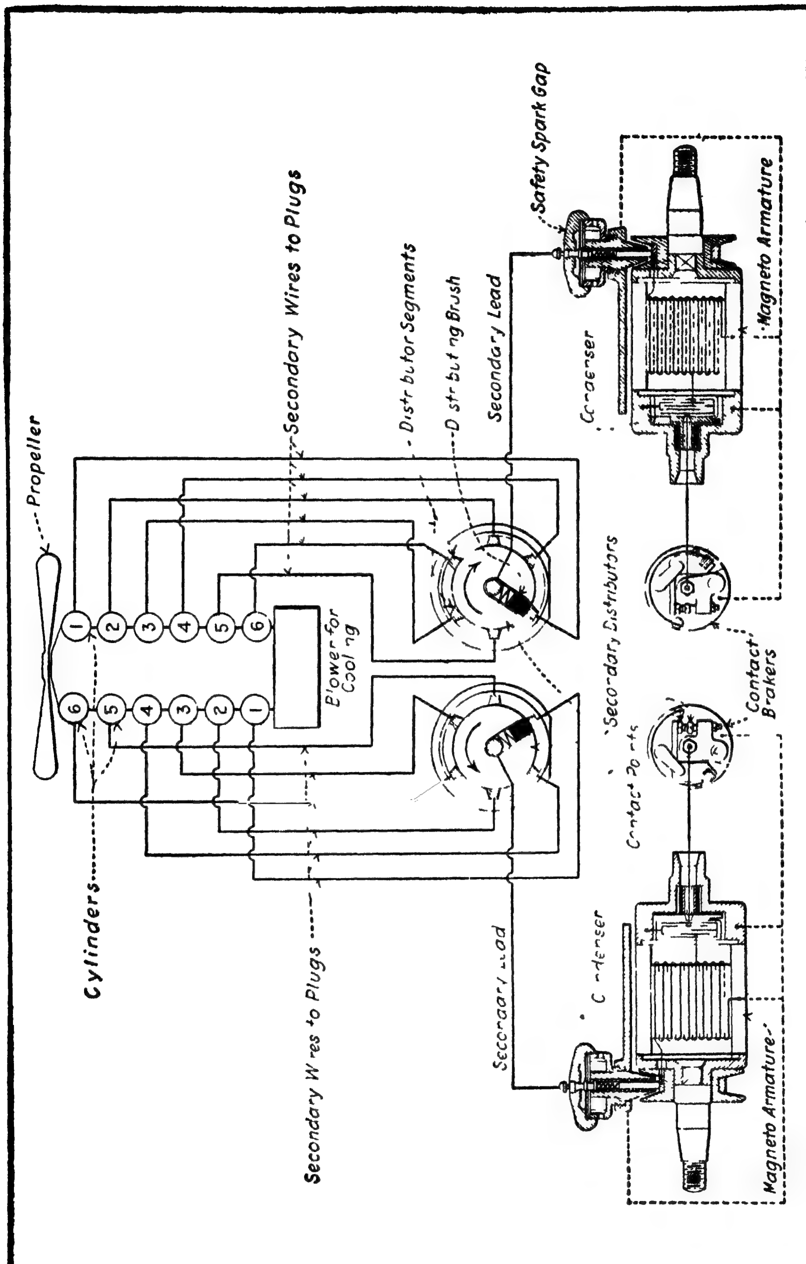


Fig. 404.—Simplified Diagram Outlining Ignition System of Renault Twelve-Cylinder Air-Cooled Aviation Engine.

at Fig. 403. The carburetor is a float feed form. Warm air is supplied for winter and damp weather by air pipes surrounding the exhaust pipes. The normal speed of the Renault engine is 1,800 r.p.m., but as the propeller is mounted upon an extension of the camshaft the normal propeller speed is but half that of the engine, which makes it possible to use a propeller of large diameter and high efficiency. Owing to the limitations of air cooling on early engines low compression was used, this being about 60 pounds per square inch, which, of course, lowers the mean effective pressure and makes the engine less efficient than water-cooled forms where it is possible to use compression pressure of 100 or more pounds per square inch. The 70 horsepower engine has cylinders with a bore of 3.78 inches and a stroke of 5.52 inches. Its weight is given as 396 pounds, when in running order, which figures 5.7 pounds per horsepower. The same cylinder size is used on the twelve-cylinder 100 horsepower and the stroke is the same. This engine in running order weighs 638 pounds, which figures approximately 6.4 pounds per brake horsepower, a figure that is more than cut in half by modern air-cooled engines of even moderate power.

QUESTIONS FOR REVIEW

1. Name some practical early aviation engine types.
2. Who was the pioneer builder of static radial engines?
3. What was the connecting rod construction of early Anzani engines?
4. What unique feature was found in early Salmson engine?
5. What types of Gnome motor were made?
6. Describe Monosoupape Gnome and state reasons why this construction is now obsolete.
7. Why do static radial and rotary single crank motors have an odd number of cylinders?
8. Describe Le Rhone rotary motor and explain how it differed from Gnome type.
9. What was the connecting rod construction of the Le Rhone motor?
10. What were the principal features of the Clerget engine?

CHAPTER XXV

TYPICAL WARTIME AVIATION ENGINES

Simplex Model A Hispano-Suiza—Early Curtiss OX Series Motor—Aeromarine Six Cylinder Vertical Motor—The Liberty Motor—Wisconsin Aviation Engines—Hall-Scott Aviation Engines—Hall-Scott Connecting Rods and Pistons—Hall-Scott Oiling—Hall-Scott Cooling System—Crankshaft and Camshaft—Mercedes Motors—Early Benz Motors—Austro-Daimler Engine—Sunbeam Aviation Engines.

A brief review of some of the wartime aviation engines is given in this chapter because some of these powerplants are still available and in use, notably the Curtiss OX5, the Hispano-Suiza and the Liberty. It is not possible to devote the necessary space to a consideration of all war time engines as details of modern forms in chapters to follow should be of greater value. The types illustrated will be of interest as comparisons may be made with engines of more recent devolpment. Instructions for repair and care of OX5 and Liberty engines will be given in proper sequence.

Simplex Model "A" Hispano-Suiza.—The Model A is of the water-cooled four-cycle Vee type, with eight cylinders, 4.7245 inch bore by 5.1182 inch stroke, piston displacement 718 cubic inches. At sea level it develops 150 horsepower at 1,450 r.p.m. It can be run successfully at much higher speeds, depending on propeller design and gearing, developing proportionately increased power. The weight, including carburetor, two magnetos, propeller hub, starting magneto and crank, but without radiator, water or oil or exhaust pipes, is 445 pounds. Average fuel consumption is .5 of a pound per horsepower hour and the oil consumption at 1,450 r.p.m. is three quarts per hour. The external appearance is shown at Fig. 405.

Four cylinders are contained in each block, which is of built-up construction; the water jackets and valve ports are cast aluminum and the individual cylinders heat-treated steel forgings threaded into the bored holes of the aluminum castings. Each block after assembly is given a number of protective coats of enamel, both inside and out, baked on. Coats on the inside are applied under pressure. The pistons are aluminum castings, ribbed. Connecting rods are tubular, of the forked type. One rod bears directly on the crankpin; the other rod has a bearing on the outside of the one first mentioned. The crankshaft is of the five-bearing type, very short, stiff in design, bored for lightness and for the oiling system. The crankshaft extension is tapered for the French standard propeller hub, which is keyed and locked to the shaft. This makes possible instant change of propellers. The case is in two halves divided on the center line of the crankshaft, the bearings being fitted between the upper and lower sections. The lower half is deep, providing a large oil reservoir and stiffening the engine. The upper half is simple and provides magneto supports on extension ledges of the two main faces. The valves are of large diameter with hollow stems, working in cast iron bushings. They are directly operated by a single hollow camshaft located over the valves. The camshafts are

driven from the crankshaft by vertical shafts and bevel gears. The camshafts, cams and heads of the valve stems are all enclosed in oil-tight removable housings of cast aluminum.

Oiling is by a positive pressure system. The oil is taken through a filter and steel tubes cast in the case to main bearings, through crankshaft to crankpins. The fourth main bearing is also provided with an oil lead from the system and through tubes running up the end of each cylinder block, oil is provided for the camshafts, cams and bearings. The surplus oil

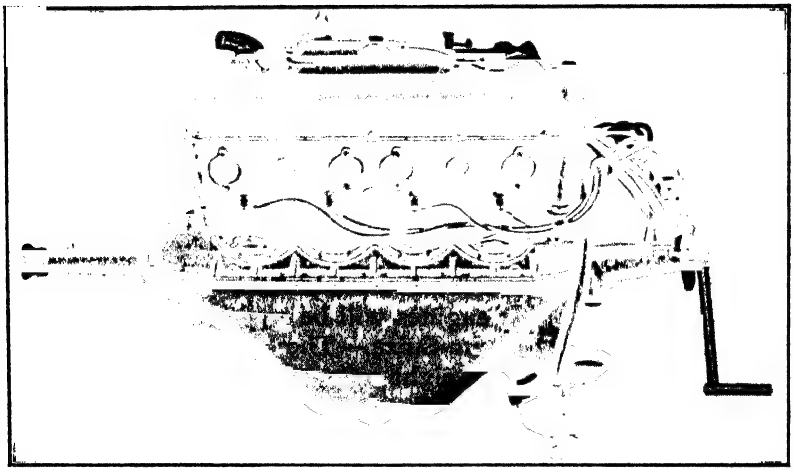


Fig. 405.—Simplex Model A Hispano-Suiza Aviation Engine was a Successful War-Time Form.

escapes through the end of the camshaft where the driving gears are mounted, and with the oil that has gathered in the top casing, descends through the drive shaft and gears to the sump. Ignition is by two eight-cylinder magnetos firing two sparkplugs per cylinder. The magnetos are driven from each of the two vertical shafts by small bevel pinions meshing in bevel gears. The carburetor is mounted between the two cylinder blocks and feeds the two blocks through aluminum manifolds which are partly water-jacketed. The engine can be equipped with a geared hand crank-starting device.

The Early Curtiss Aviation Motors.—The Curtiss OX motor has eight cylinders, four inch bore, five inch stroke, delivers 90 horsepower at 1,400 turns, and the weight turns out at 4.17 pounds per horsepower. This motor has cast iron cylinders with monel metal jackets, overhead inclined valves operated by means of two rocker arms, push-and-pull rods from the central camshaft located in the crankcase. The cam and push rod design is extremely ingenious and the whole valve construction turns out very light. This motor is an evolution from the early Curtiss type motor which was used by Glenn Curtiss when he won the Gordon Bennett Cup at Rheims. A slightly larger edition of this type motor is the OXX5, as shown at Figs. 406 and 407, which has cylinders 4¼ inches by five inches, delivers 100 horsepower at 1,400 turns and has the same fuel and oil consumption

as the OX type motor, namely, 60 pound of fuel per brake horsepower hour and .03 pound of lubricating oil per brake horsepower hour.

The Curtiss Company also developed a larger-sized motor known as the V2, which was originally rated at 160 horsepower and which has since been refined and improved so that the motor gives 220 horsepower at 1,400 turns, with a fuel consumption of $^{52}/_{100}$ of a pound per brake horsepower hour and an oil consumption of 02 of a pound per brake horsepower hour This larger motor has a weight of 3 45 pounds per horsepower and is said to have given very satisfactory service. The V2 motor has drawn steel

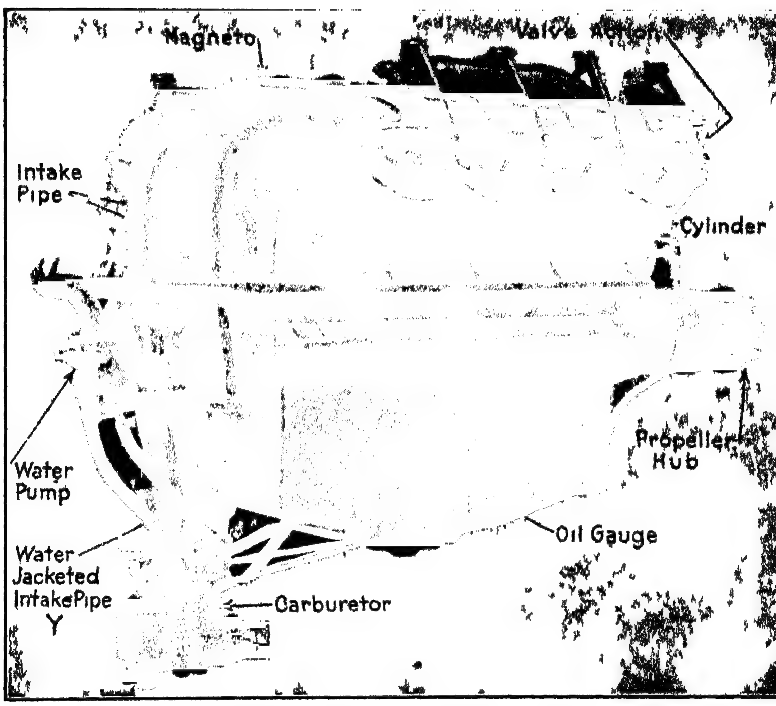


Fig. 406.—Curtiss OX5 Water-Cooled Aviation Engine Was an Eight-Cylinder Type Largely Used in War-Time Training Machines and Still Used in Various Forms of Moderate Weight Commercial Airplanes.

cylinders, with a bore of five inches and a stroke of seven inches, with a steel water jacket top and a monel metal cylindrical jacket, both of which are brazed on to the cylinder barrel itself. Both these motors use side by side connecting rods and fully forced lubrication. The camshafts act as a gallery from which the oil is distributed to the camshaft bearings, the main crankshaft bearings, and the gearing Here again we find extremely short rods, which, as before mentioned, enables the height and the consequent weight of construction to be very much reduced. For ordinary flying at altitudes of 5,000 to 6,000 feet, the motors are sent out with an aluminum liner, bolted between the cylinder and the crankcase in order to give a compression ratio which does not result in preignition at a low altitude.

For high flying, however, these aluminum liners were taken out and the compression volume is decreased to about 18.6 per cent of the total volume.

The Curtiss Airplane Company also built a twelve-cylinder five inch by seven inch motor, which was designed for aeronautical uses primarily. This engine was rated at 250 horsepower, but it was said to develop 300 at 1,400 r.p.m. Weights—Motor, 1,125 pounds; radiator, 120 pounds; cooling water, 100 pounds; propeller, 95 pounds. Gasoline consumption

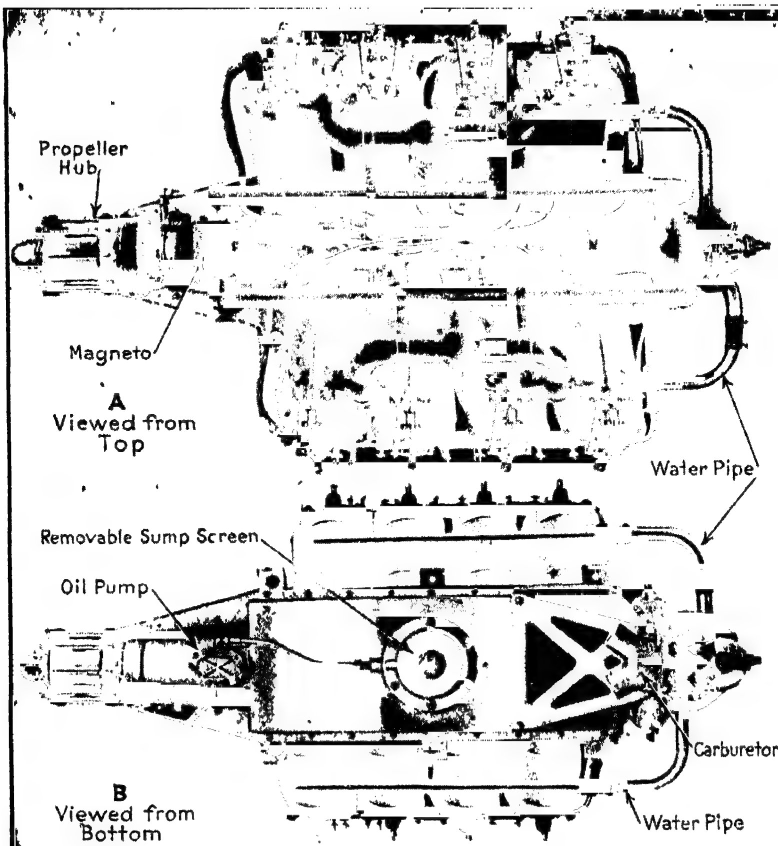


Fig. 407.—Top and Bottom Views of Curtiss OX5 100 Horsepower Aviation Engines.

per horsepower hour, $\frac{6}{10}$ pounds. Oil consumption per hour at maximum speed—two pints. Installation dimensions—Overall length, $84\frac{5}{8}$ inches; overall width, $34\frac{1}{8}$ inches; overall depth, 40 inches; width at bed, $30\frac{1}{2}$ inches; height from bed, $21\frac{1}{8}$ inches; depth from bed, $18\frac{1}{2}$ inches.

Aeromarine Six-Cylinder Vertical Motor.—These motors are four-stroke cycle, six-cylinder vertical type, with cylinder $4\frac{5}{16}$ inch bore by $5\frac{1}{8}$ inch stroke. The general appearance of this motor is shown in illustration at Fig. 408. This engine is rated at 85-90 horsepower. All reciprocating and revolving parts of this motor are made of the highest grades of steel

obtainable as are the studs, nuts and bolts. The upper and lower parts of crankcase are made of composition aluminum casting. Lower crankcase is made of high grade aluminum composition casting and is bolted directly to the upper half. The oil reservoir in this lower half casting provides sufficient oil capacity for five hours' continuous running at full power. Increased capacity can be provided if needed to meet greater endurance requirements. Oil is forced under pressure to all bearings by means of

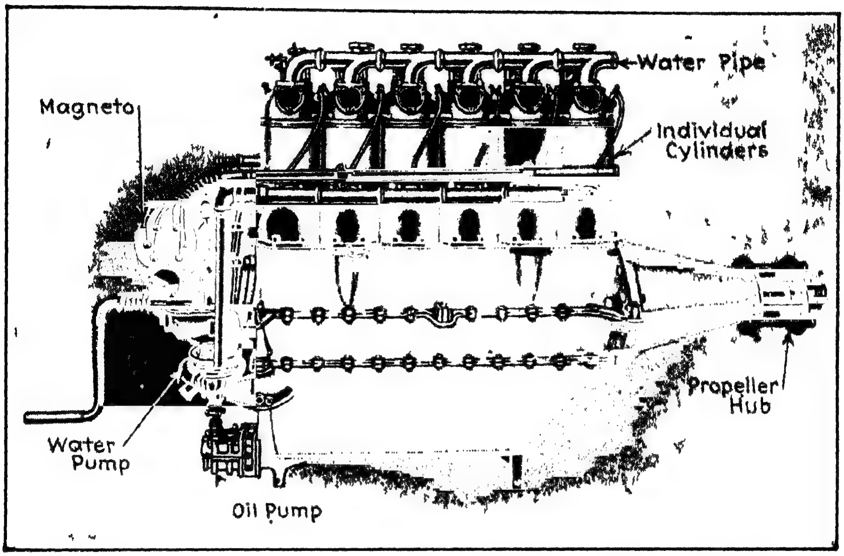


Fig. 408.—The Early Six-Cylinder Aeromarine Aviation Engine was a Successful War-Time Type that Received Considerable Application in Small Flying Boats.

high-pressure duplex-geared pumps One side of this pump delivers oil under pressure to all the bearings, while the other side draws the oil from the splash case and delivers it to the main sump. The oil reservoir is entirely separate from the crankcase chamber. Under no circumstances will oil flood the cylinder, and the oiling system is not affected in any way by any angle of flight or position of motor. An oil pressure gauge is placed on instrument board of machine, which gives at all times the pressure in oil system, and a sight glass at lower half of case indicates the amount of oil contained. The oil pump is external on magneto end of motor, and is very accessible. An external oil strainer is provided, which is removable in a few minutes' time without the loss of any oil. All oil from reservoir to the motor passes through this strainer. Pressure gauge feed is also attached and can be piped to any part of machine desired.

The cylinders are made of high-grade castings and are machined and ground accurately to size. Cylinders are bolted to crankcase with chrome nickel steel studs and nuts which securely lock cylinder to upper half of crankcase. The main retaining cylinder studs go through crankcase and support crankshaft bearings so that crankshaft and cylinders are tied together as one unit. Water jackets are of copper, $\frac{1}{16}$ of an inch thick,

electrically deposited. This makes a noncorrosive metal. Cooling is furnished by a centrifugal pump, which delivers 25 gallons per minute at 1,400 r.p.m. Pistons are made of cast iron, accurately machined and ground to exact dimensions, which are carefully balanced. Piston rings are semi-steel rings of Aeromarine special design.

Connecting rods are of chrome nickel steel, H-section. Crankshaft is made of chrome nickel steel, machined all over, and cut from solid billet, and is accurately balanced through the medium of balance weights being

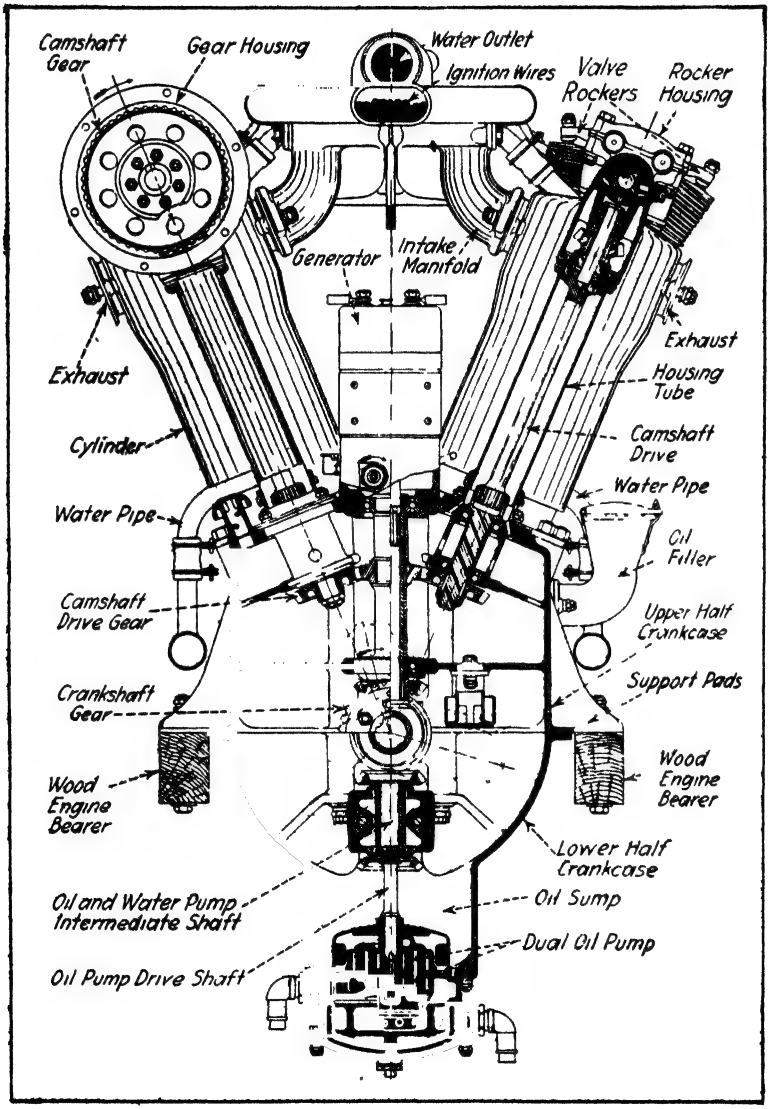


Fig. 409.—Views Showing Construction of the Liberty Aviation Engine. Note Method of Oil Pump and Camshaft Drive.

forged integral with crank. It is drilled for lightness and plugged for force feed lubrication. There are seven main bearings to crankshaft. All bearings are of high-grade babbitt, die cast, and are interchangeable and easily replaced. The main bearings of the crankshaft are provided with a single groove to take oil under pressure from pressure tube which is cast integral with case. Connecting rod bearings are of the same type. The gudgeon pin is hardened, ground and secured in connecting rod, and is allowed to work in piston. Camshaft is of steel, with cams forged integral, drilled for lightness and forced-feed lubrication, and is case-hardened. The bearings of camshaft are of bronze. Magneto, two high-tension Bosch D.U. 6. The intake manifold for carburetors are aluminum castings and are so designed that each carburetor feeds three cylinders, thereby insuring easy flow of vapor at all speeds. Weight, 420 pounds.

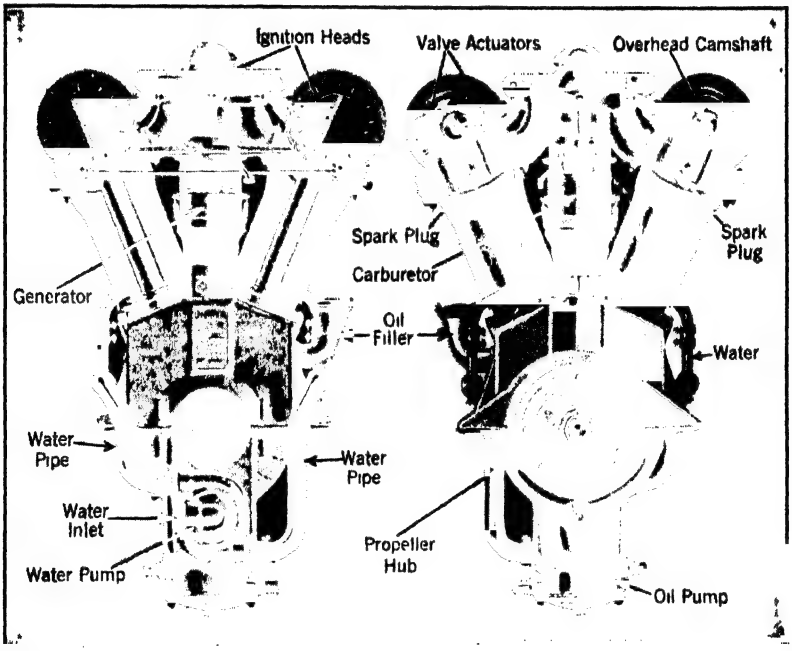


Fig. 410.—Rear and Front Views of the Liberty Aviation Engine. An American Development that Was Produced in Large Quantities in 1918 but Which is Now Obsolescent.

The Liberty Motor.—This very practical powerplant was designed for the equipment division of the Signal Corps, United States Army, by a commission of leading engineers working under the direction of Major J. G. Vincent, Chief Engineer of the Packard Motor Car Company and Major E. J. Hall, of the Hall-Scott Motor Car Company shortly after our entry into the World War. The object was to design a standard engine that could be put into quantity production and built by the same methods that were applied to the production of automobiles in motor car plants. Many thousands of these motors were built to interchangeable standards. There

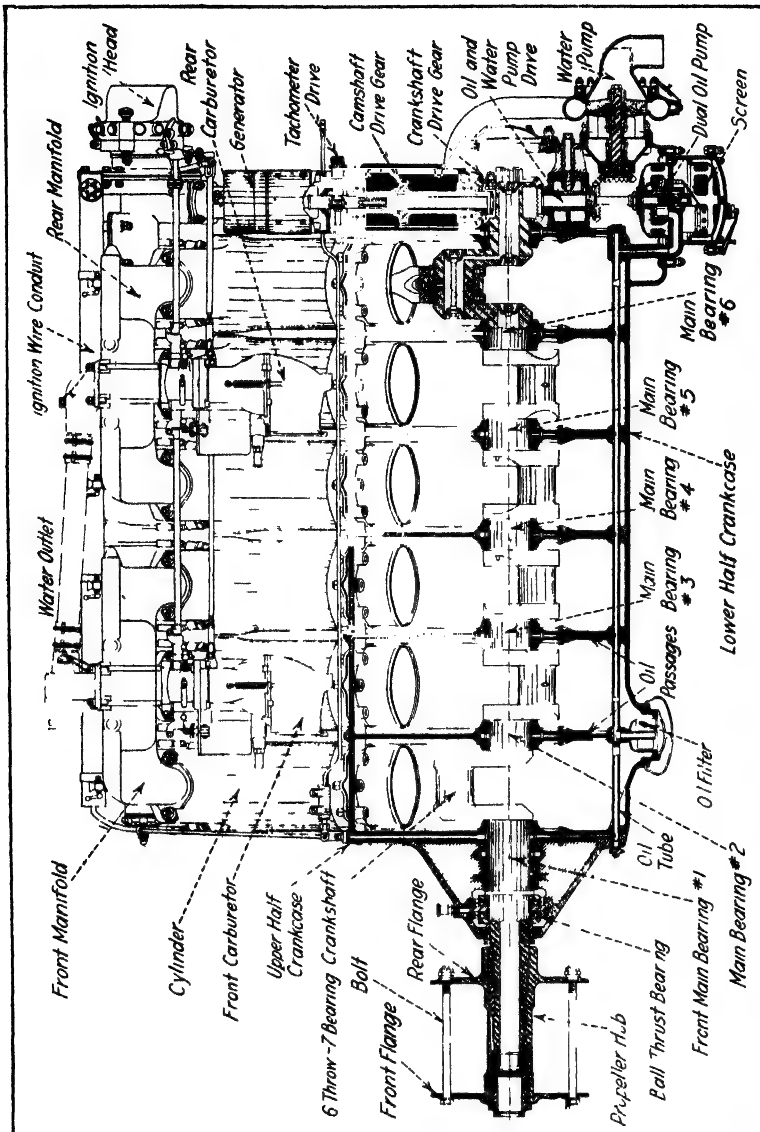


Fig. 411.—Longitudinal Section of Liberty Engine Crankcase Showing Simplicity and Strength of Parts.

has probably been no motor that was criticized as much or as unjustly as this one. It proved to be a very practical and reliable type in service when compared to contemporary designs of foreign manufacture. Designers of more recently developed types take great pleasure in pulling this design to pieces and showing its weak points when compared to the newer engines without taking into consideration that if it was not for the experience gained with this and other early engines that the modern highly refined powerplants would not have been possible. Such comparisons are not fair, and when viewed in the light of the knowledge that obtained when this

engine was first designed; it will always remain an outstanding achievement of American engineering and productive skill. The Liberty engine construction can be understood by referring to illustrations Figs. 409 and 410 which show external views and sectional drawings Figs. 411 and 412 inclusive which show mechanical details. The cylinders are 45 degrees apart. The cylinder bore is five inches, the stroke is seven inches. The cubic displacement is 1,650 cubic inches. The horsepower is 400 at 1,700 r.p.m. The compression ratio is 5.40 to 1 and a mean effective pressure of 113 pounds per square inch is obtained. The engine weighs 806 pounds, as shipped, which gives a dry weight of slightly more than two pounds per horsepower.

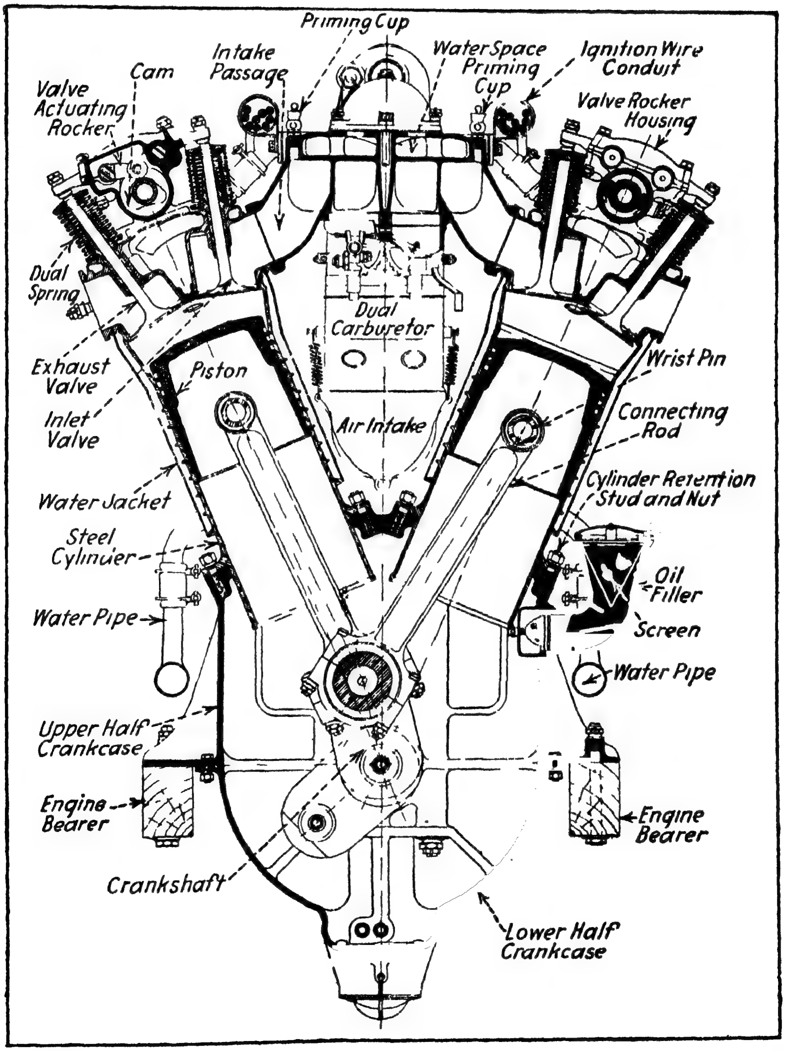


Fig. 412—Transverse Section of Liberty Engine Showing Method of Valve Actuation and Internal Arrangement of Parts.

The water pump water passages and cylinder jackets from face of pump inlet to the face of the water outlet hold 5.5 gallons or 46 pounds of water. The fuel-consumption is .54 pounds per horsepower hour or 36 gallons per hour with wide open throttle at 1,700 r.p.m. The oil consumption is .03 pounds per horsepower hour or 1.5 gallons per hour with wide open throttle. Sufficient radiator capacity should be provided to hold the water temperature at not to exceed 200 degrees Fahrenheit and the water temperature should not be allowed to become lower than 160 degrees Fahrenheit or carburetion troubles will result. Ignition is by special Delco battery system. Two Dual Zenith carburetors furnish the mixture.

The valves are actuated by overhead camshafts driven by bevel gearing and vertical shafts. The ignition distributors are mounted at the rear ends of the camshafts. The cylinders are steel with applied and welded sheet steel jackets. The crankcase is aluminum alloy, made in two pieces and is divided on the vertical center line of the crankshaft. The crankshaft is a six-throw seven main bearing type. The connecting rods are of the scissors type, two rods acting on one crankpin.

Pistons are of aluminum alloy, having three wide grooves above the wristpin, each groove being fitted with one ring. Each piston is provided with seven circumferential oil distributing grooves and the piston is relieved around the wristpin bosses. Two forms of pistons are available, a flat top for low compression or training and Navy engines and a domed top for high compression types. The oiling system is a pressure feed dry sump type, and has been fully described in chapter on lubrication, the internal parts of the cylinder being lubricated by the oil spray thrown off centrifugally by the revolving crankshaft.

Air-Cooled Liberty Engine.—The air-cooled Liberty engine shown at Fig. 464 A is a "Vee" type air-cooled engine obtained by substituting air-cooled cylinders for the original jacketed cylinders. It differs from the water-cooled form only in the cylinder, piston, induction system, and valve gear assemblies. The crankcase, crankshaft, connecting rods and main accessory drive train, are the same as similar parts of the water-cooled engine. The bore was reduced to 4⅝ inches in the air-cooled cylinders thus providing cooling space between them and reducing the piston displacement to 1,411 cubic inches. The crankcase is of cast-aluminum in two sections parted at the crankshaft centerline, the upper half carrying the cylinders and the main bearing upper halves; the lower half supporting the main bearing lower halves just as in the water-cooled engine. The crankshaft is a steel forging drilled and plugged to provide oil passages. The propeller hub is mounted on a taper and located by a key in the usual manner. The main accessory drive gear is bolted to a flange at the rear end of the shaft. The forged steel connecting rods are of "H" section, the secondary rod mounted on the center of the main rod bearing and working directly upon the bronze shell of the bearing. The pistons are die cast aluminum alloy with three rings above the piston pin and one in the skirt. The cylinders have steel barrels with integral fins screwed and shrunk in a cast aluminum-alloy head. A steel ring is shrunk on the aluminum head below the lowest fin to clamp the head to the barrel. The combustion-chamber top is hemispherical with bronze valve seats shrunk

in. The valves are tulip shaped; the exhaust valve is cooled internally by a partial filling of sodium and potassium nitrate according to the practice fully described in another chapter. One intake and one exhaust valve are used in each cylinder. They are operated by rocker arms from a single camshaft on each bank. The valve mechanism is fully enclosed and runs in a bath of oil. The fuel mixture for the engine is supplied by a single Stromberg NA-S8J carburetor through a gear driven low altitude supercharger at the rear of the engine. Ignition is by the standard Liberty-Delco system.

The air-cooled Liberty engine has been built in upright and inverted types and with both geared and direct drive. With a compression ratio of 5.4:1 it develops 436 horsepower at 1,900 r.p.m. The brake mean effective pressure at that output is 128 pounds per square inch. The weight of the ungeared engine is 1,010 pounds, (2.32 pounds per horsepower), nearly 200 pounds lighter than the standard Liberty engine with radiator and cooling water.

Wisconsin Aviation Engines.—The six-cylinder Wisconsin aviation engines, one of which is shown at Fig. 413, were of the vertical type, with cylinders in pairs and valves in the head. The cylinders were made of aluminum alloy castings, bored and machined and then fitted with hardened steel sleeves about 1/16 of an inch thickness. After these sleeves had been shrunk into the cylinders, they were finished by grinding in place. Gray iron valve seats are cast into the cylinders. The valve seats and cylinders, as well as the valve ports, are entirely surrounded by water jackets. The valves set in the heads at an angle of 25 degrees from the vertical, are made of tungsten steel and are provided with double springs, the outer or main spring and the inner or auxiliary spring, which is used as a precautionary measure to prevent a valve falling into the cylinder in remote case of a main spring breaking. The camshaft is made of one solid forging, case-hardened. It is carried in an aluminum housing bolted to the top of the cylinders. This housing is split horizontally, the upper half carrying the chrome vanadium steel rocker levers. The lower half has an oil return trough cast integral, into which the excess oil overflows and then drains back to the crankcase. Small inspection plates are fitted over the cams and inner ends of the cam rocker levers. The camshaft runs in bronze bearings and the drive is through vertical shaft and bevel gears.

The crankcase was made of aluminum, the upper half carrying the bearings for the crankshaft. The lower half carries the oil sump in which all of the oil except that circulating through the system at the time is carried. The crankshaft is made of chrome vanadium steel of an elastic limit of 115,000 pounds. The crankpins and ends of the shaft are drilled for lightness and the cheeks are also drilled for oil circulation. The crankshaft runs in bronze-backed, Fahrig metal-lined bearings, four in number. A double thrust bearing is also provided, so that the motor may be used either in a tractor or pusher type of machine. Outside of the thrust bearing an annular ball bearing is used to take the radial load of the propeller. The propeller is mounted on a taper. At the opposite end of the shaft a bevel gear is fitted which drives the camshaft, through a vertical shaft, and also drives the water and oil pumps and magnetos. All gears are made of

chrome vanadium steel, heat-treated.

The connecting rods are tubular and machined from chrome vanadium steel forgings. Oil tubes are fitted to the rods which carry the oil up to the wristpins and pistons The rods complete with bushings weigh 5½ pounds each. The pistons are made of aluminum alloy and are very light

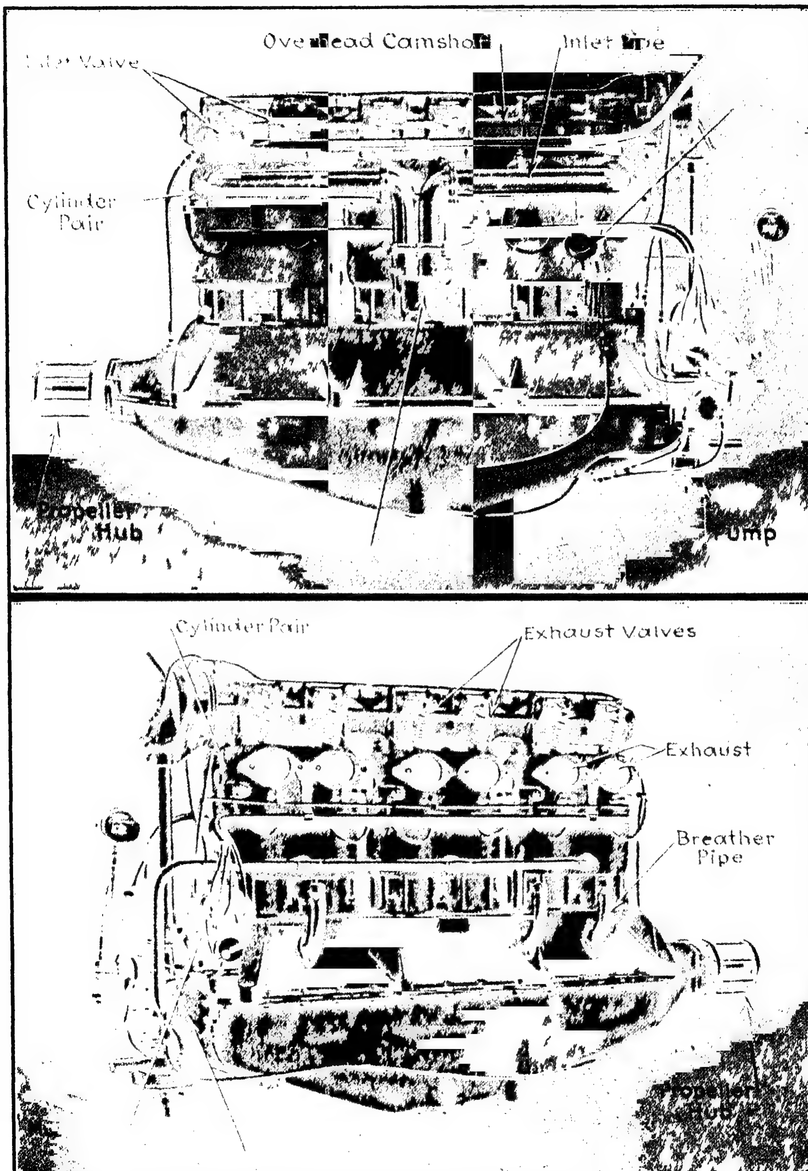


Fig. 413.—The Wisconsin Aviation Engine, a War-Time Form of Excellent Design, as Viewed from the Carburetor Side at the Top and from the Exhaust Side at the Bottom of the Illustration.

and strong, weighing only two pounds two ounces each. Two leak-proof rings are fitted to each piston. The wristpins are hollow, of hardened steel, and are free to turn either in the piston or the rod. A bronze bushing is fitted in the upper end of the rod, but no bushing is fitted in the pistons, the hardened steel wristpins making an excellent bearing in the aluminum alloy.

The water circulation is by centrifugal pump, which is mounted at the lower end of the vertical shaft. The water is pumped through brass pipes to the lower end of the cylinder water jackets and leaves the upper end of the jackets just above the exhaust valves. The lubricating system is one of the main features of the engines, being designed to work with the motor at any angle. The oil is carried in the sump, from where it is taken by the oil circulating pump through a strainer and forced through a header, extending the full length of the crankcase, and distributed to the main bearings. From the main bearings it is forced through the hollow crankshaft to the connecting rod big ends and then through tubes on the rods to wristpins and pistons. Another lead takes oil from the main header to the camshaft bearings. The oil forced out of the ends of the camshaft bearings fills pockets under the cams and in the cam rocker levers. The excess flows back through pipes and through the train of gears to the crankcase. A strainer is fitted at each end of the crankcase, through which the oil is drawn by separate pumps and returned to the sump. Either one of these pumps is large enough to take care of all of the return oil, so that the operation is perfect whether the motor is inclined up or down. No splash is used in the crankcase, the system being a full force feed. An oil level indicator is provided, showing the amount of oil in the sump at all times. The oil pressure in these motors is carried at ten pounds, a relief valve being fitted to hold the pressure constant.

Ignition is by two Bosch magnetos, each on a separate set of plugs fired simultaneously on opposite sides of the cylinders. Should one magneto fail, the other would still run the engine at only a slight loss in power. The Zenith double carburetor is used, three cylinders being supplied by each carburetor. This insures a higher volumetric efficiency, which means more power, as there is no over-lapping of inlet valves whatever by this arrangement. All parts of these motors are very accessible. The water and oil pumps, carburetors, magnetos, oil strainer or other parts can be removed without disturbing other parts. The lower crankcase can be removed for inspection or adjustment of bearings, as the crankshaft and bearing caps are carried by the upper half. The motor supporting lugs are also part of the upper crankcase.

The six-cylinder motor, without carburetors or magnetos, weighs 547 pounds. With carburetor and magnetos, the weight is 600 pounds. The weight of cooling water in the motor is 38 pounds. The sump will carry four gallons of oil, or about 28 pounds. A radiator can be furnished suitable for the motor, weighing 50 pounds. This radiator will hold three gallons of water or about 25 pounds. The motor will drive a two-blade, eight feet diameter by 6.25 feet pitch Paragon propeller 1,400 revolutions per minute, developing 148 horsepower. The weight of this propeller is 42 pounds. This makes a total weight of motor, complete with propeller, radiator filled with water, but without lubricating oil, 755 pounds, or about

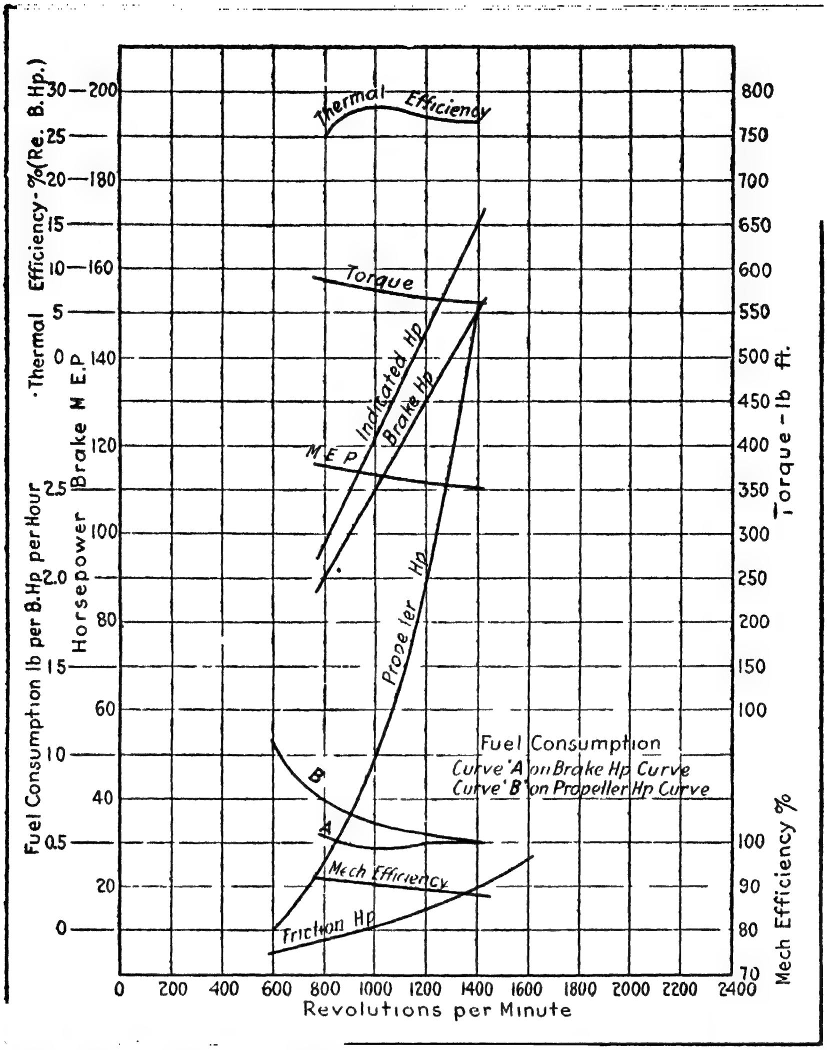


Fig. 414.—Power, Torque and Efficiency Curves of Early Wisconsin Aviation Motor, Presented so Comparison can be Made with Performance of Modern Forms.

5.1 pounds per horsepower for complete powerplant. The fuel consumption is .5 pound per horsepower per hour. The lubricating oil consumption is .0175 pound per horsepower per hour, or a total of 2.6 pounds per hour at 1,400 revolutions per minute. This would make the weight of fuel and oil, per hour's run at full power at 1,400 revolutions per minute, 76.6 pounds.

Following are the principal dimensions of the six-cylinder motor:

Bore 5 inches.
Stroke 6½ inches.
Crankshaft diameter throughout 2 inches.

Length of crankpin and main bearings 3½ inches.
Diameter of valves 3 inches (2¾ inches clear).
Lift of valves ½ inch.
Volume of compression space 22 per cent of total.
Diameter of wristpins 1$\frac{3}{16}$ inches.
Firing order 1, 4, 2, 6, 3, 5.

The horsepower developed at 1,200 revolutions per minute is 130, at 1,300 revolutions per minute 140, at 1,400 revolutions per minute 148. 1,400 is the maximum speed at which it is recommended to run these motors.

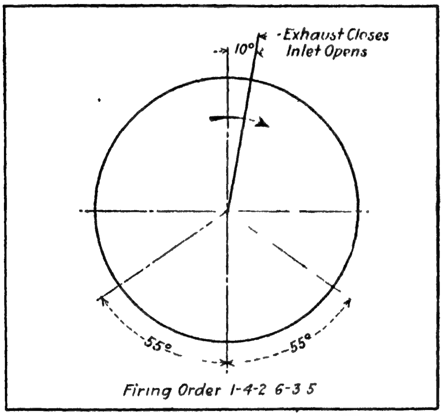


Fig. 415.—Timing Diagram Showing Firing Order and Valve Timing of Wisconsin Aviation Engine.

A twelve-cylinder Vee-type engine was also built by this company, similar in dimensions of cylinders to the six. The principal differences were in the drive to camshaft, which is through spur gears instead of bevel. A hinged type of connecting rod was used which did not increase the length of the motor and, at the same time, this construction provided for ample bearings. A double centrifugal water pump was provided for this motor, so as to distribute the water uniformly to both sets of cylinders. Four magnetos were used, two for each set of six cylinders. The magnetos were very accessibly located on a bracket on the spur gear cover. The carburetors were located on the outside of the motors, where they are very accessible, while the exhaust is in the center of the valley. The crankshaft on the twelve is 2½ inches in diameter and the shaft is bored to reduce weight.

Hall-Scott Aviation Engines.—The following specifications of the Hall-Scott "Big Four" engines apply just as well to the six-cylinder vertical types which are practically the same in construction except for the structural changes necessary to accommodate the two extra cylinders. Cylinders are cast separately from a special mixture of semi-steel, having cylinder head with valve seats integral. Special attention has been given to the design of the water jacket around the valves and head, there being two inches of water space above same. The cylinder is annealed, rough machined, then the inner cylinder wall and valve seats ground to mirror finish. This adds to the durability of the cylinder, and diminishes a great deal of the excess friction.

Great care is taken in the casting and machining of these cylinders, to have the bore and walls concentric with each other. Small ribs are cast between outer and inner walls to assist cooling as well as to transfer stresses direct from the explosion to hold-down bolts which run from steel main bearing caps to top of cylinders. The cylinders are machined upon

the sides so that when assembled on the crankcase with grooved hold-down washers tightened, they form a solid block, greatly assisting the rigidity of crankcase.

Hall-Scott Connecting Rods.—The connecting rods are very light, being of the I beam type, milled from a solid chrome nickel die forging. The caps are held on by two ½-inch twenty-thread chrome nickel through bolts. The rods are first roughed out, then annealed. Holes are drilled, after which the rods are hardened and holes ground parallel with each other. The piston end is fitted with a gun metal bushing, while the crankpin end

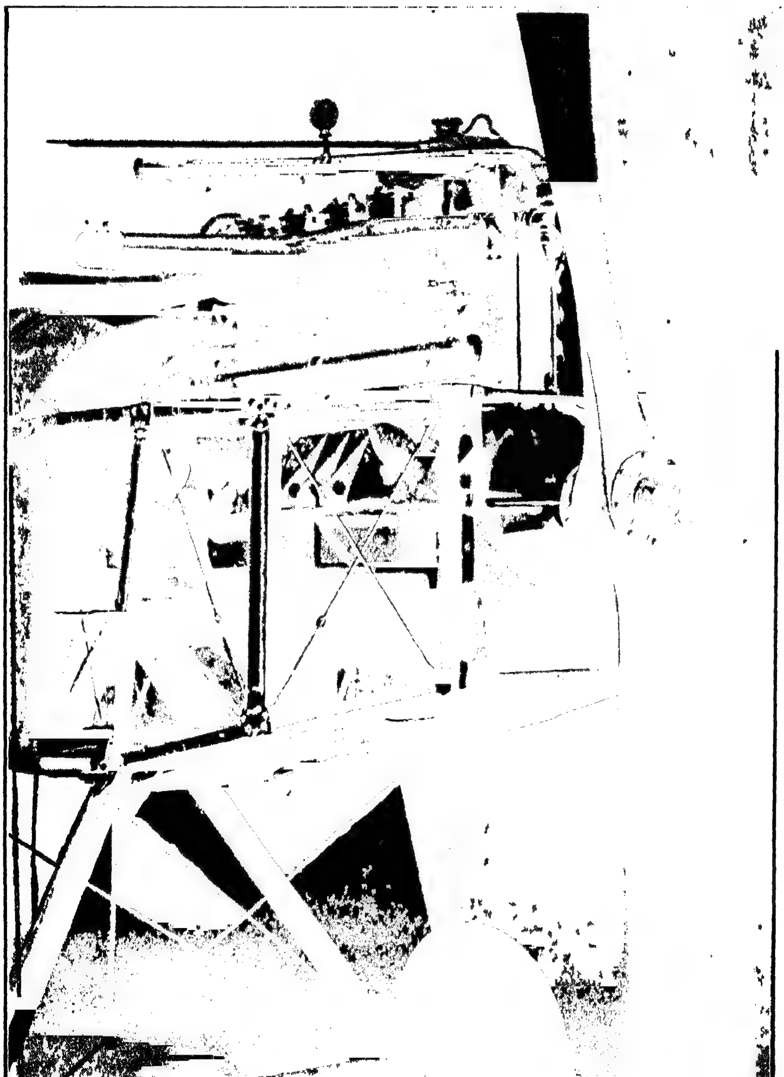


Fig. 416.—Front View of Early Biplane Fuselage Showing Installation of Hall-Scott Six-Cylinder Aviation Engine with Direct Driven Tractor Screw which Turns at Engine Speed.

carries two bronze serrated shells, which are tinned and babbitted hot, being broached to harden the babbitt. Between the cap and rod proper are placed laminated shims for adjustment.

Hall-Scott Oiling.—The oiling system is known as the high pressure type, oil being forced to the under side of the main bearings with from five to 30 points pressure. This system is not affected by extreme angles obtained in flying, or whether the motor is used for push or pull machines. A large gear pump is located in the lowest point of the oil sump, and being submerged at all times with oil, does away with troublesome stuffing boxes and check valves. The oil is first drawn from the strainer in oil sump to the long jacket around the intake manifold, then forced to the main distributor pipe in crankcase, which leads to all main bearings. A bypass, located at one end of the distributor pipe, can be regulated to provide any pressure required, the surplus oil being returned to the case. A special feature of this system is the dirt, water and sediment trap, located at the bottom of the oil sump. This can be removed without disturbing or dismantling the oil pump or any oil pipes. A small oil pressure gauge is provided, which can be run to the aviator's instrument board. This registers the oil pressure, and also determines its circulation.

Hall-Scott Cooling System.—The cooling of this motor is accomplished by the oil as well as the water, this being covered by patent No. 1,078,919. This is accomplished by circulating the oil around a long intake manifold jacket; the carburetion of gasoline cools this regardless of weather conditions. Crankcase heat is therefore kept at a minimum. The uniform temperature of the cylinders is maintained by the use of ingenious internal outlet pipes, running through the head of each of the six cylinders, rubber hose connections being used so that any one of the cylinders may be removed without disturbing the others. Slots are cut in these pipes so that cooler water is drawn directly around the exhaust valves. Extra large water jackets are provided upon the cylinders, two inches of water space is left above the valves and cylinder head. The water is circulated by a large centrifugal pump insuring ample circulation at all speeds.

Crankshaft and Camshaft.—The crankshaft is of the five bearing type, being machined from a special heat treated drop forging of the highest grade nickel steel. The forging is first drilled, then roughed out. After this the shaft is straightened, turned down to a grinding size, then ground accurately to size. The bearing surfaces are of extremely large size, oversize, considering general practice in the building of high-speed engines of similar bore and stroke. The crankshaft bearings are two inches in diameter by $1\frac{15}{16}$ of an inch long, excepting the rear main bearing, which is $4\frac{3}{8}$ inches long, and front main bearing, which is $2\frac{3}{16}$ inches long. Steel oil scuppers are pinned and sweated onto the webs of the shaft, which allows of properly oiling the connecting rod bearings. Two thrust bearings are installed on the propeller end of the shaft, one for pull and the other for push. The propeller is driven by the crankshaft flange, which is securely held in place upon the shaft by six keys. These drive an outside propeller flange, the propeller being clamped between them by six through bolts. The flange is fitted to a long taper on crankshaft. This enables the propeller to be removed without disturbing the bolts. Timing gears and

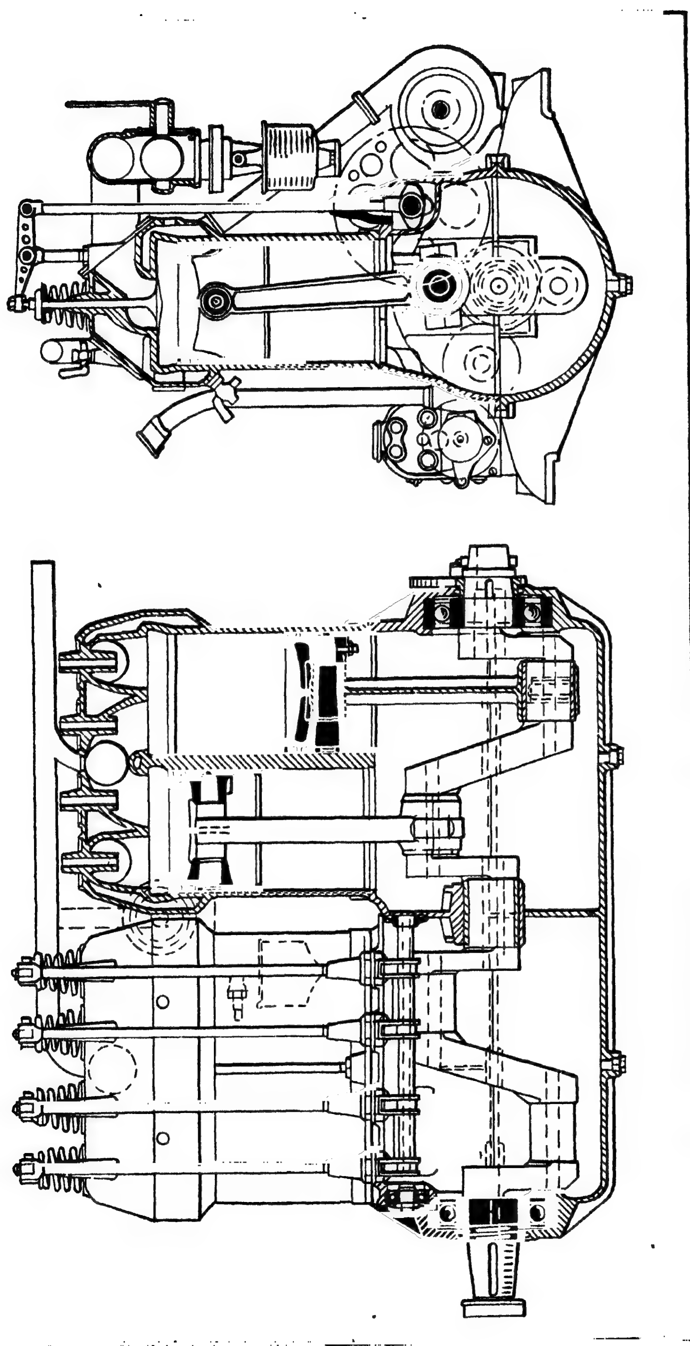


Fig. 417.—Side and End Sectional Views of Four-Cylinder Argus Engine, a German 100 Horsepower War-Time Design, Having a Bore and Stroke of 5.60 Inches, and Developing its Power at 1,368 R.P.M. This Engine Weighed Three and One-Half Pounds per Horsepower Dry.

starting ratchets are bolted to a flange turned integral with shaft.

The camshaft is of the one piece type, air pump eccentric, and gear flange being integral. It is made from a low carbon specially heat treated nickel forging, is first roughed out and drilled entire length; the cams are then formed, after which it is case hardened and ground to size. The camshaft bearings are extra long, made from Parson's white brass. A small clutch is milled in gear end of shaft to drive revolution indicator. The camshaft is enclosed in an aluminum housing bolted directly on top of all six cylinders, being driven by a vertical shaft in connection with bevel gears. This shaft, in conjunction with rocker arms, rollers and other working parts, is oiled by forcing the oil into end of shaft, using same as a distributor, allowing the surplus supply to flow back into the crankcase through hollow vertical tube. This supply oils the magneto and pump gears. Extremely large Tungsten valves, being one-half the cylinder diameter, are seated in the cylinder heads. Large diameter oil tempered springs held in tool steel cups, locked with a key, are provided. The ports are very large and short, being designed to allow the gases to enter and exhaust with the least possible resistance. These valves are operated by overhead one piece camshaft in connection with short chrome nickel rocker arms. These arms have hardened tool steel rollers on cam end with hardened tool steel adjusting screws opposite. This construction allows accurate valve timing at all speeds with least possible weight.

Crankcases are cast of the best aluminum alloy, hand scraped and sand blasted inside and out. The lower oil case can be removed without breaking any connections, so that the connecting rods and other working parts can readily be inspected. An extremely large strainer and dirt trap is located in the center and lowest point of the case, which is easily removed from the outside without disturbing the oil pump or any working parts.

A Zenith carburetor is provided. Automatic valves and springs are absent, making the adjustment simple and efficient. This carburetor is not affected by altitude to any appreciable extent. A Hall-Scott device, covered by U. S. Patent No. 1,078,919, allows the oil to be taken direct from the crankcase and run around the carburetor manifold, which assists carburetion as well as reduces crankcase heat. Two waterproof four-cylinder Splitdorf "Dixie" magnetos are provided. Both magneto interruptors are connected to a rock shaft integral with the motor, making outside connections unnecessary. It is worthy of note that with this independent double magneto system, one complete magneto can become inoperative, and still the motor will run and continue to give good power.

The pistons as provided in the A7 engines are cast from a mixture of steel and gray iron. These are extremely light, yet provided with six deep ribs under the arch head, greatly aiding the cooling of the piston as well as strengthening it. The piston pin bosses are located very low in order to keep the heat from the piston head away from the upper end of the connecting rod, as well as to arrange them at the point where the piston fits the cylinder best. Three ¼ inch rings are carried. The pistons as provided in the A7a engines are cast from aluminum alloy. Four ¼ inch rings are carried. In both piston types a large diameter, heat treated, chrome nickel steel wristpin is provided, assembled in such a way as to

assist the circular rib between the wristpin bosses to keep the piston from being distorted from the explosions.

Mercedes Motor.—The 150 horsepower six-cylinder Mercedes motor is 140 millimeters bore and 160 millimeters stroke. The Merecedes company started with smaller-sized cylinders, namely 100 millimeters bore and 140 millimeters stroke, six-cylinders. The principal features of the design are forged steel cylinders with forged steel elbows for gas passages, pressed steel water jackets, which when welded together forms the cylinder assembly, the use of inclined overhead valves operated by means of an overhead camshaft through rocker arms which multiply with the motion of the

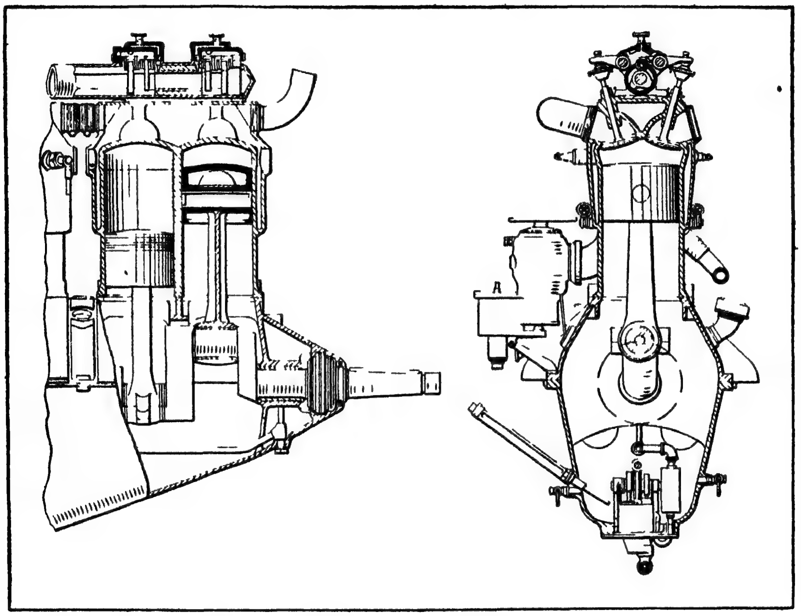


Fig. 418.—Part Sectional View of 90 Horsepower Mercedes Engine, an Early German Design Typical Also of the Larger Sizes.

cam. By the use of steel cylinders, not only is the weight greatly reduced, but certain freedom from distortion through unequal sections, leaks and cracks are entirely avoided. The construction is necessarily very expensive. It is certainly a sound job. In the details of this construction there are a number of important things, such as finished gas passages, water-cooled valve guides and a very small mass of metal, which is water-cooled, surrounding the sparkplug. Of course, it is necessary to use very high compression in aviation motors in order to secure high power and economy and owing to the fact that aviation motors are worked at nearly their maximum, the heat flow through the cylinder, piston, and valves is many times higher than that encountered in automobile motors. It has been found necessary to develop special types of pistons to carry the heat from the center of the head in order to prevent preignition. In the Mercedes motor the pistons have a drop forged steel head which includes the piston

boss and this head is screwed into a cast-iron skirt which has been machined inside to secure uniform wall thickness.

The carburetor used on this 150 horsepower Mercedes motor is precisely of the same type used on the twin six motor. It has two venturi throats, in the center of which is placed the gasoline spray nozzle of conventional type, fixed size orifices, immediately above which are placed two panel type throttles with side outlets. An idling or primary nozzle is arranged to discharge above the top of the venturi throat. The carburetor body is of cast aluminum and is water-jacketed. It is bolted directly to air passage passing through the top and bottom half of the crankcase which passes down through the oil reservoir. The air before reaching the carburetor proper to some extent has cooled the oil in the crank chamber and has itself been heated to assist in the vaporization. The inlet pipes themselves are copper. All the passages between the venturi throat and the inlet valve have been carefully finished and polished. The only abnormal thing in the design of this motor is the short connecting rod which is considerably less than twice the stroke and would be considered very bad practice in motor car engines. A short connecting rod, however, possesses two very real virtues in that it cuts down height of the motor and the piston passes over the bottom dead center much more slowly than with a long rod.

Other features of the design are a very stiff crankcase, both halves of which are bolted together by means of long through bolts, the crankshaft main bearings are seated in the lower half of the case instead of in the usual caps and no provision is made for taking up the main bearings. The Mercedes company uses a plunger type of pump having mechanically operated piston valves and it is driven by means of worm gearing.

The overhead camshaft construction is extremely light. The camshaft is mounted in a nearly cylindrical cast bronze case and is driven by means of bevel gears from the crankshaft. The vertical bevel gear shaft through which the drive is taken from the crankshaft to the camshaft operates at one and one-half times the crankshaft speeds and the reduction to the half-time camshaft is secured through a pair of bevels. On this vertical shaft there is mounted the water pump and a bevel gear for driving two magnetos. The water pump mounted on this shaft tends to steady the drive and avoid vibration in the gearing.

The cylinder sizes of six-cylinder aviation motors which have been built by Mercedes are:

Bore	Stroke	Horsepower
105 mm.	140 mm.	100
120 mm.	140 mm.	135
140 mm.	150 mm.	150
140 mm.	160 mm.	160

The largest of these early motors had its horsepower increased to 176 at 1,450 r.p.m. This general design of motor has been the foundation for a great many other aviation motor designs, some of which have proved very successful but none of which is equal to the original. Among the motors which follow more or less closely the scheme of design and arrange-

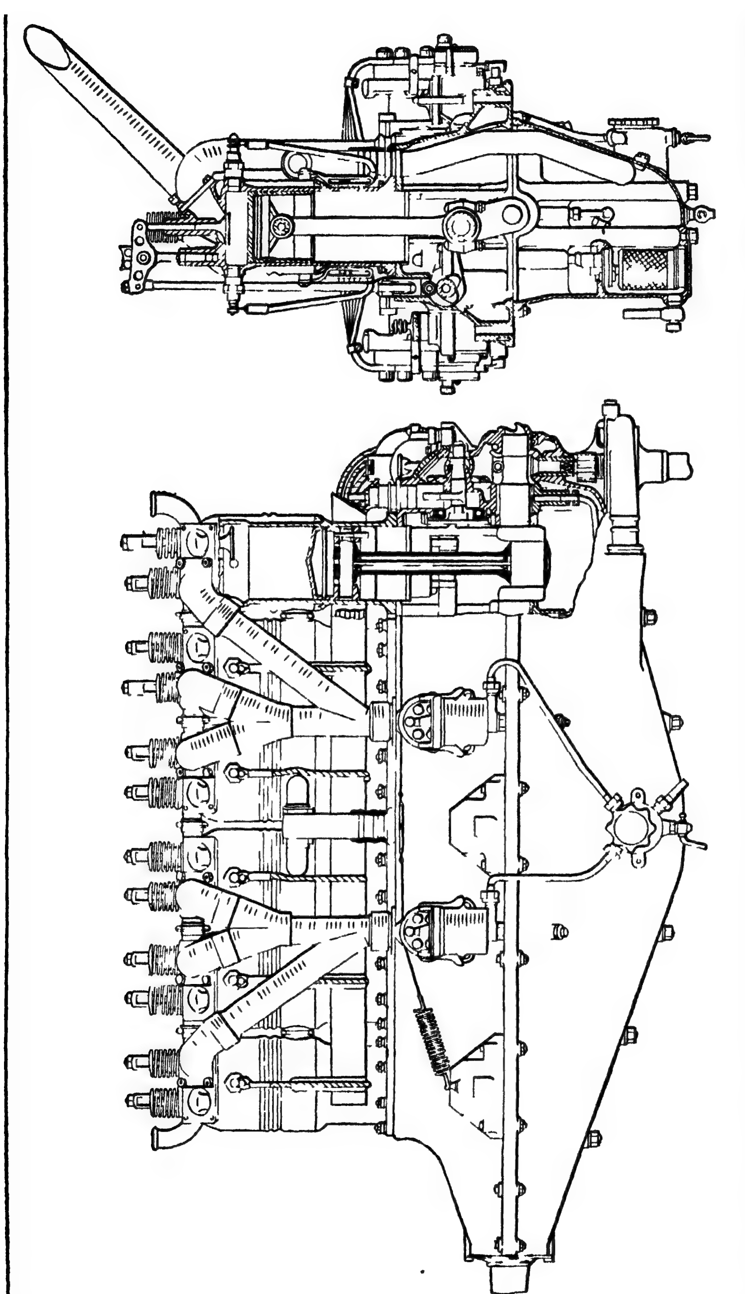


Fig. 419.—Part Sectional Side View and Sectional End View of the Benz 160 Horsepower Aviation Engine, which was Used Widely in German War-Time Airplanes.

ment were the Hall-Scott, the Wisconsin motor, the Renault water-cooled, the Packard, the Christofferson and the Rolls-Royce. Each of these motors shows considerable variation in detail. The Rolls-Royce and Renault are the only ones who have used the steel cylinder with the steel jacket. The Wisconsin motor uses an aluminum cylinder with a hardened steel liner and cast-iron valve seats. The Christofferson has somewhat similar design to the Wisconsin with the exception that the valve seats are threaded into the aluminum jacket and the cylinder head has a blank end which is secured to the aluminum casting by means of the valve seat pieces. The Rolls-Royce motors showed small differences in details of design in cylinder head and camshaft housing from the Mercedes on which it had taken out patents, not only abroad but in this country.

The Benz Motor.—In the Kaiser prize contest for aviation motors a four-cylinder Benz motor of 130 by 180 millimeters won first prize, developing 103 brake horsepower at 1,290 r.p.m. The fuel consumption was 210 grams per horsepower hour. Total weight of the motor was 153 kilograms. The oil consumption was .02 of a kilogram per horsepower hour. This motor was afterward expanded into a six-cylinder design and three different sizes were built.

The accompanying table gives some of the details of weight, horsepower, etc.

Motor Type	B	FD	FF
Rated horsepower	85	100	150
Horsepower at 1250 r.p.m.	88	108	150
Horsepower at 1350 r.p.m.	95	115	160
Bore in millimeters	106	116	130
Stroke in millimeters	150	160	180
Offset of the cylinders in millimeters	18	20	20
Rate of gasoline consumption in grams	240	230	225
Oil consumption in grams per b.h.p. hour	10	10	10
Oil capacity in kilograms	36	4	4½
Water capacity in litres	5½	7½	9½
The weight with water and oil but with two magnetos, fuel feeder and air pump in kilograms	170	200	245
The weight of motors, including the water pump, two magnetos, double ignition, etc.	160	190	230
The weight of the exhaust pipe, complete in kilograms	4	4.8	5½
The weight of the propeller hub in kilograms	3½	4	4

The Benz cylinder is a simple, straightforward design and a very reliable construction and not particularly difficult to manufacture. The cylinder is cast of iron without a water jacket but including 45 degrees angle elbows to the valve ports. The cylinders are machined wherever possible and at other points have been hand filed and scraped, after which a jacket, which is pressed in two halves, is gas welded by means of short pipes welded on to the jacket. The bottom and the top of the cylinders become water galleries, and by this means separate water pipes with their attendant weight and complication are eliminated. Rubber rings held in aluminum clamps serve to connect the cylinders together. The whole construction turns out very neat and light. The cylinder walls are four

millimeters or $\frac{3}{16}$ of an inch thick and the combustion-chamber is of cylindrical pancake form and is 140 millimeters or 5.60 of an inch in diameter. The valve seats are 68 millimeters in diameter and the valve port is 62 millimeters in diameter.

The passage joining the port is 57 millimeters in diameter. In order to insert the valves into the cylinder the valve-stem is made with two diameters and the valve has to be cocked to insert it in the guide, which has a bronze bushing at its upper end to compensate for the smaller valve stem diameter. The valve stem is fourteen millimeters or $\frac{9}{16}$ of an inch in diameter and is reduced at its upper portion to 9½ millimeters. The valves are operated through a push rod and rocker arm construction, which is $\frac{7}{16}$ of an inch and exceedingly light. Rocker arm supports are steel studs with enlarged heads to take a double row ball bearing. A roller is mounted at one end of the rocker arm to impinge on the end of the valve-stem, and the rocker arm has an adjustable globe stud at the other end. The push rods are light steel tubes with a wall thickness of 0.75 millimeters and have a hardened steel cup at their upper end to engage the rocker arm globe stud and a hardened steel globe at their lower end to socket in the roller plunger.

The Benz camshaft has a diameter of 26 millimeters and is bored straight through eighteen millimeters and there is a spiral gear made integrally with the shaft in about the center of its length for driving the oil pump gear. The cam faces are ten millimeters wide. There is also, in addition to the intake and exhaust cams, a set of half compression cams. The shaft is moved longitudinally in its bearings by means of an eccentric to put these cams into action. At the fore end of the shaft is a driving gear flange which is very small in diameter and very thin. The flange is 68 millimeters in diameter and four millimeters thick and is tapped to take six millimeter bolts. The total length of camshaft is 1,038 millimeters, and it becomes a regular gun boring job to drill a hole of this length.

The camshaft gear is 140 millimeters or 5½ inches outside diameter. It has 54 teeth and the gear face is fifteen millimeters or $\frac{19}{32}$ of an inch. The flange and web have an average thickness of four millimeters or $\frac{5}{32}$ of an inch and the web is drilled full of holes interposed between the spur gear mounted on the camshaft and the camshaft gear. There is a gear which serves to drive the magnetos and tachometer, also the air pump. The shaft is made integrally with this gear and has an eccentric portion against which the air pump roll plunger impinges.

The seven-bearing crankshaft is finished all over in a beautiful manner, and the shaft out of the particular motor we have shows no signs of wear whatever. The crankpins are 55 millimeters in diameter and 69 millimeters long. Through both the crankpin and main bearings there is drilled a 28 millimeter hole, and the crank cheeks are plugged with solder. The crank cheeks are also built to convey the lubricant to the crankpins. At the fore end of the crank cheek there is pressed on a spur driving gear. There is screwed on to the front end of the shaft a piece which forms a bevel water pump driving gear and the starting dog. At the rear end of the shaft very close to the propeller hub mounting there is a double thrust bearing to take the propeller thrust.

Long, shouldered studs are screwed into the top half of the crankcase portion of the case and pass clean through the bottom half of the case. The case is very stiff and well ribbed. The three center bearing diaphragms have double walls. The center one serves as a duct through which water pipe passes, and those on either side of the center form the carburetor intake air passages and are enlarged in section at one side to take the carburetor barrel throttle.

The pistons are of cast iron and carry three concentric rings ¼ of an inch wide on their upper end, which are pinned at the joint. The top of the piston forms the frustum of the cone and the pistons are 110 millimeters in length. The lower portion of the skirt is machined inside and has a wall thickness of one millimeter. Riveted to the piston head is a conical diaphragm which contacts with the piston pin when in place and serves to carry the heat off the center of the piston.

The oil pump assembly comprises a pair of plunger pumps which draw oil from a separate outside pump, and constructed integrally with it is a gear pump which delivers the oil under about 60 pounds pressure through a set of copper pipes in the base to the main bearings. The plunger oil pump shows great refinement of detail. A worm wheel and two eccentrics are machined up out of one piece and serve to operate the plungers.

Some interesting details of the 160 horsepower Benz motor, show how carefully the design had been considered.

Maximum horsepower, 167.5 brake horsepower.

Speed at maximum horsepower, 1,500 r.p.m.

Piston speed at maximum horsepower, 1,770 feet per minute.

Normal horsepower, 160 brake horsepower.

Speed at normal horsepower, 1,400 r.p.m.

Piston speed at normal horsepower, 1,656 feet per minute.

Brake mean pressure at maximum horsepower, 101.2 pound per square inch.

Brake mean pressure at normal horsepower, 103.4 pound per square inch.

Specific power cubic inch swept volume per brake horsepower, 5.46 cubic inches; 160 brake horsepower.

Weight of piston, complete with gudgeon pin, rings, etc., 5.0 pound.

Weight of connecting rod, complete with bearings, 4.99 pound; 1.8 pound reciprocating.

Weight of reciprocating parts per cylinder, 6.8 pound.

Weight of reciprocating parts per square inch of piston area, 0.33 pound.

Outside diameter of inlet valve, 68 millimeters; 2.68 inches.

Diameter of inlet valve port (d), 61.5 millimeters; 2.42 inches.

Maximum lift of inlet valve (h), eleven millimeters, 0.443 inch.

Area of inlet valve opening (π d h), 21.25 square centimeters; 3.29 square inches.

Inlet valve opens, degrees on crank, top dead center.

Inlet valve closes, degrees on crank, 60 degrees late; 35 millimeters late.

Outside diameter of exhaust valve, 68 millimeters, 2.68 inches.

Diameter of exhaust valve port (d), 61.5 millimeters; 2.42 inches.

Maximum lift of exhaust valve (h) eleven millimeters; 0.433 inch.

Area of exhaust valve opening (π d h), 21.25 square centimeters; 3.29 square inches.

Exhaust valve opens, degrees on crank, 60 degrees early; 35 millimeters early.

Exhaust valve closes, degrees on crank, 16½ degrees late; five millimeters late.

Length of connecting rod between centers, 314 millimeters; 12.36 inches.

Ratio connecting rod to crank throw, 3.49:1.

Diameter of crankshaft, 56 millimeters outside, 2.165 inches; 28 millimeters inside, 1.102 inches.

Diameter of crankpin, 55 millimeters outside, 2.165 inches; 28 millimeters inside, 1.102 inches.

Diameter of gudgeon pin, 30 millimeters outside, 1.181 inches; nineteen millimeters inside, 0.708 inch.

Diameter of camshaft, 26 millimeters outside, 1.023 inches; eighteen millimeters inside, 0.708 inch.

Number of crankshaft bearings, seven.

Projected area of crankpin bearings, 36.85 square centimeters; 5.72 square inches.

Projected area of gudgeon pin bearings, 22.20 square centimeters; 3.44 square inches.

Firing sequence, 1, 5, 3, 6, 2, 4.

Type of magnetos, ZH6 Bosch.

Direction of rotation of magneto from driving end, one clock, one anticlock.

Magneto timing, full advance, 30 degrees early (sixteen millimeters early).

Type of carburetors (2) Benz design.

Fuel consumption per hour, normal horsepower, 0.57 pint.

Normal speed of propeller, engine speed, 1,400 r.p.m.

Austro-Daimler Engine.—One of the first very successful European flying engines which was developed in Europe is the Austro-Daimler, which is shown in end section in a preceding chapter. The first of these motors had four-cylinders, 120 by 140 millimeters, bore and stroke, with cast-iron cylinders, overhead valves operated by means of a single rocker arm, controlled by two cams and the valves were closed by a single leaf spring which oscillates with the rocker arm. The cylinders are cast singly and have either copper or steel jackets applied to them. The four-cylinder design was afterwards expanded to the six-cylinder design and still later a six-cylinder motor of 130 by 175 millimeters was developed. This motor used an offset crankshaft, as did the Benz motor, and the effect of offset has been discussed earlier in this treatise. The Benz motor also uses an offset camshaft which improves the valve operation and changes the valve lift diagram. The lubrication also is different than any other aviation motor, since individual high pressure metering pumps are used to deliver fresh oil only to the bearings and cylinders, as was the custom in automobile practice some fifteen years ago.

Sunbeam Aviation Engines.—These very successful engines have been developed by Louis Coatalen. At the opening of the war the largest size Coatalen motor was 225 horsepower and was of the L-head type having a single camshaft for operating valves and was an evolution from the

Characteristics of Typical American Pre-War Aviation Engines

Maker's Name and Model	Number of Cyl.	Bore (Inches)	Stroke (Inches)	Piston Displacement (Cubic Inches)	H.P.	R.P.M.	Weight of Engine with Carburetor and Ignition	Gas Consumption
Curtiss OX	8	4	5	502.6	90	1400	375	10 gals. per hour
Curtiss OXX2	8	4¼	5	567.5	100	1400	423	11 gals. per hour
Curtiss V2	8	5	7	1100	200	1400	690	....
Duesenberg A4	4	4¾	7	496	140	2100	455	....
General Vehicle Gnome Mono (Rotary Air-Cooled)	9	4.33	5.9	848	100	1200	272	12 gals. per hour at rated hp.
Hall-Scott A7	4	5	7	550	90–100	1400	410	....
Hall-Scott A5	6	5	7	825	125	1300	592	....
Hispano Suiza	8	4⅝	5	672	154	1500	455	....
Sturtevant 5A	8	4	5½	....	140	2000	514	13.75 gals. per hour
Thomas 8	8	4	5½	552.9	135	2000	630 with self-starter	0.59 lbs. per B. H. P. hr.
Thomas 88	8	4⅛	5½	552.9	150	2100	525 lbs. with self-starter	0.59 lbs. per B. H. P. hr.
Wisconsin	6	5	6½	765.7	140	1380	637	....

(Note.—Engines running at speeds in excess of 1500 R.P.M. have a reduction gear for driving propeller.)

twelve-cylinder racing car which the Sunbeam Company had previously built. Since 1914 the Sunbeam Company have produced engines of six-, eight-, twelve-, and eighteen-cylinders from 150 to 500 horsepower with both iron and aluminum cylinders. The war-time motors had overhead camshafts with a separate shaft for operating the intake and exhaust valves. Camshafts are connected through to the crankshaft by means of a train of spur gears, all of which are mounted on two double row ball bearings.

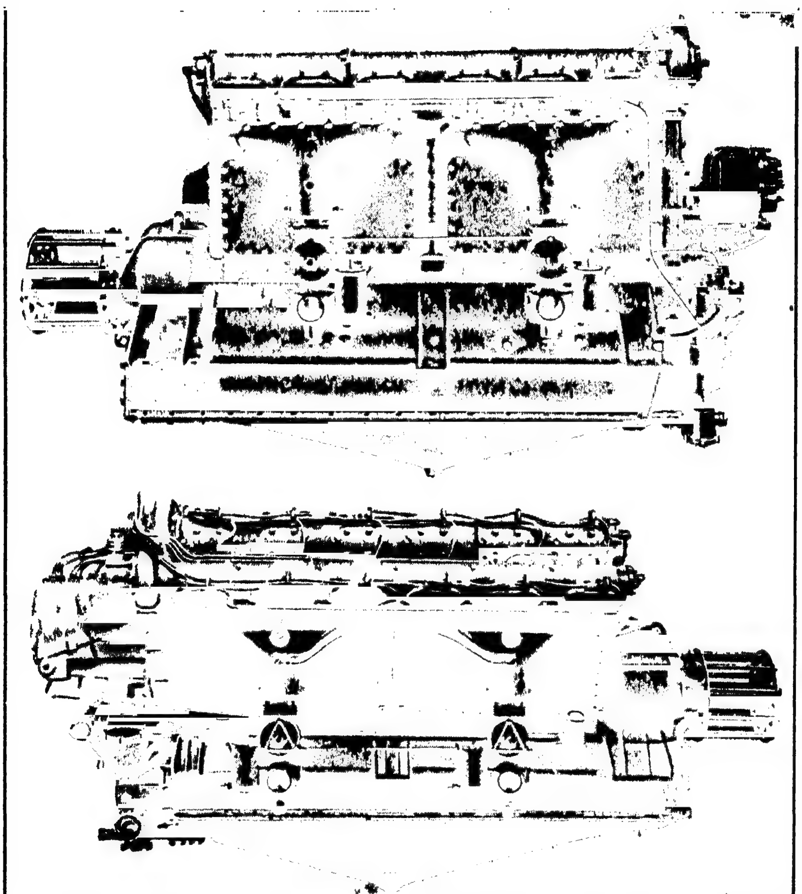


Fig. 420.—At Top, Sunbeam Overhead Valve 170 Horsepower Six-Cylinder Water-Cooled Engine. The Side View Below the Six-Cylinder is of the Sunbeam 350 Horsepower Twelve-Cylinder, "Vee" Engine.

In the twin six, 350 horsepower engine, operating at 2,100 r.p.m., requires about four horsepower to operate the camshafts. This motor gives 362 horsepower at 2,100 revolutions and has a fuel consumption of $^{51}/_{100}$ of a pint per brake horsepower hour. The cylinders are 110 by 160 millimeters. The same design has been expanded into an eighteen-cylinder which gives 525 horsepower at 2,100 turns There has also been developed a very successful eight-cylinder motor rated at 2,220 horsepower which has

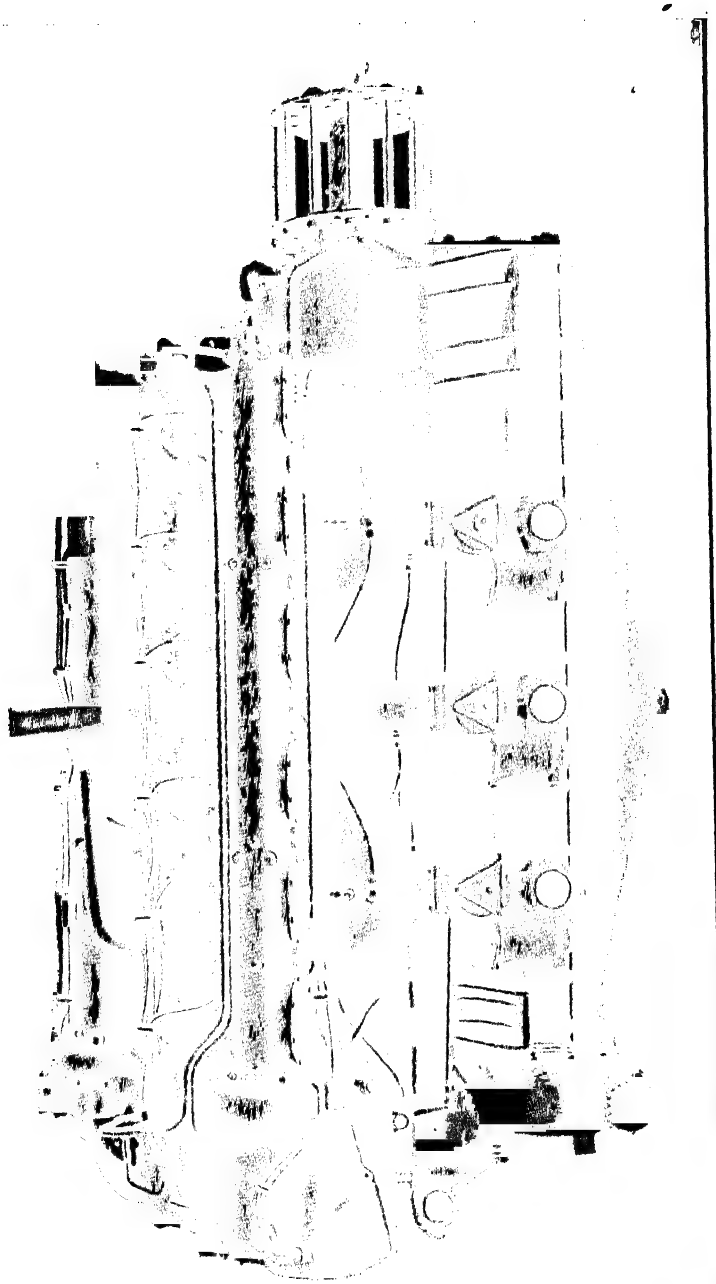


Fig. 421.—View of the Eighteen-Cylinder "W" Type Sunbeam-Coatalen Aircraft Engine Rated at 475 Brake Horsepower. This Engine Had Extremely Good Weight to Horsepower Ratio and Was a Remarkably Efficient War-Time Design.

a bore and stroke of 120 by 130 millimeters, weight 450 pounds. This motor is an aluminum block construction with steel sleeves inserted. Three valves are operated, one for the inlet and two for the exhaust. One camshaft operates the three valves.

The wartime Sunbeam engines operated with a mean effective pressure of 135 pounds with a compression ratio of six to one sea level and compared favorably in performance with engines built today. The connecting rods are of the articulated type as in the Renault motor and are very short. The

Fig. 422.—Sunbeam Eighteen-Cylinder Motor, Viewed from the Anti-Propeller End, Showing Water Pump and Ignition Magneto Installation.

weight of these motors turns out at 2 6 pounds per brake horsepower, and they are able to go through a 100 hour test without any trouble of any kind. The lubricating system comprises a dry base and oil pump for drawing the oil off from the base, whence it is delivered to the filter and cooling system. It then is pumped by a separate high pressure gear pump through the entire motor. In these larger European motors, castor oil is used largely for lubrication. It is said that without the use of castor oil it is impossible to hold full power for five hours. Coatalen favors aluminum cylinders rather than cast iron. The series of views in Figs. 420 to 423 inclusive, illustrates the vertical, narrow type of engine; the Vee-form; and the broad arrow type wherein three rows, each of six cylinders, are set on a common crankcase. In this water-cooled series the gasoline and oil consumption are notably low, as is the weight per horsepower.

In the eighteen-cylinder overhead valve Sunbeam-Coatalen aircraft engine of 475 brake horsepower, there are no fewer than half a dozen magnetos. Each magneto is inclosed. Two sparks are furnished to each cylinder from independent magnetos. On this engine there are also no fewer than six carburetors. Shortness of crankshaft, and therefore of engine length, and absence of vibration are achieved by the linking of the connecting rods. Those concerned with three--cylinders in the broad arrow formation work on one crankpin, the outer rods being linked to the central master one. In consequence of this arrangement, the piston travel in the case of the

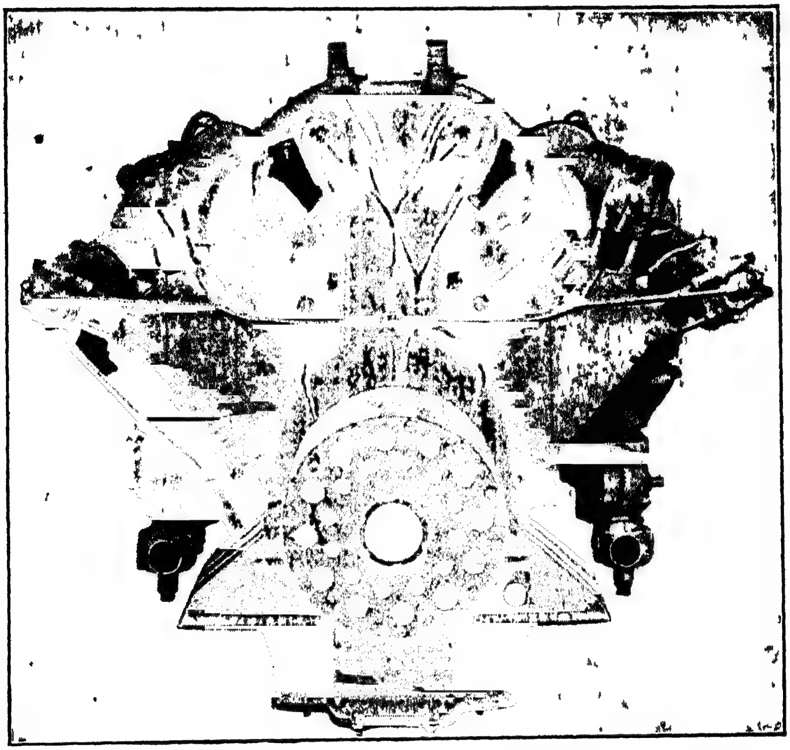


Fig. 423.—Propeller End of the Sunbeam Eighteen-Cylinder 475 Brake Horsepower Aviation Engine.

central row of cylinders is 160 millimeters, while the stroke of the pistons of the cylinders set on either side is in each case 168 millimeters. Inasmuch as each set of six-cylinders is completely balanced in itself, this difference in stroke does not affect the balance of the engine as a whole. The duplicate ignition scheme also applies to the twelve-cylinder 350 brake horsepower Sunbeam-Coatalen overhead valve aircraft engine type. It is distinguishable, incidentally, by the passage formed through the center of each induction pipe for the sparking plug in the center cylinder of each block of three. In this, as in the eighteen-cylinder and the six-cylinder types, there are two camshafts for each set of cylinders. These camshafts are lubricated by low pressure and are operated through a train of inclosed spur

wheels at the magneto end of the machine. The six-cylinder, 170 brake horsepower vertical type employs the same general principles, including the detail that each carburetor serves gas to a group of three-cylinders only. It will be observed that this engine presents notably little head resistance, being suitable for multi-engined aircraft.

THE FOREMOST ENGINES OF 1918

Engines	Symbol	No. of Cyls.	Hp.	R.P.M.	Weight	Lbs. Hp.	B.M.E.P.	Comp. Ratio	Cooling and Type
Liberty	12a	12	400	1700	845	2.11	119.5	5.5 -1	W. C. Vee
Hispano	41	8	180	1700	485	2.69	124.5	5.3 -1	W. C. Vee
Hispano	H	8	300	1600	596	1.98	127.8	5.3 -1	W. C. Vee
Mercedes		6	160	1250	618	3.86	105.2	4.8 -1	W. C. Vertical
Mercedes		6	260	1400	935	3.75	107.5	4.94-1	W. C. Vertical
Maybach		6	200	1200	900	4.5	107.4	4.95-1	W. C. Vertical
Benz		6	220	1400	829	3.68	113.0	4.93-1	W. C. Vertical
Rolls-Royce	Falcon	12	220	2000	723	2.68	113.0	5.3 -1	W. C. Vee
Rolls-Royce	Eagle	12	360	1800	933	2.66	127.5	5.3 -1	W. C. Vee
Fiat	A12	6	300	1600	928	3.32	112.0	4.31-1	W. C. Vertical
Lorraine-Dietrich		8	275	1600	546	1.98	137.0	4.7 -1	W. C. Vertical
Renault		12	300	1550	836	2.78	116.0	2.66-1	W. C. Vee
A. B. C.	Wasp	7	170	1750	260	1.53	111.2	4.0 -1	A. C. Radial
A. B. C.	Dragon-Fly	9	320	1650	600	1.88	103.5	4.0 -1	A. C. Radial
LeRhone	J	9	110	1200	320	2.91	87.5	4.82-1	A. C. Rotary
Bentley	BR2	9	200	1300	475	2.0	77.3	5.3 -1	A. C. Rotary
Salmson		9	250	1500	473	1.89	100.0	5.4 -1	W. C. Radial
Curtiss	OX5	8	90	1400	390	4.33	111.0	4.92-1	W. C. Vee

QUESTIONS FOR REVIEW

1. What were the features of the Model A Hispano-Suiza motor?
2. What was the most popular motor for aviation training purposes and what were its important features?
3. Describe Liberty motor construction.
4. What changes were made in the Liberty engine to adapt it to air cooling?
5. Describe construction of Hall-Scott engines.
6. Do late types of aircraft engines differ materially from the early models and if so, in what way?
7. Name two popular German war-time engines.
8. Outline some features of the Benz engine.

CHAPTER XXVI

INSTALLATION AND TROUBLE SHOOTING OF WARTIME ENGINES

Early Inverted Engine Mounting—Conventional Installation—Mounting of Curtiss OX Series Engines—Engine Bed Dimensions—Hall-Scott Engine Installation—Fuel Supply System—Ignition Switches—Hall-Scott Water Systems—Preparations to Start Engine—Installing Early Rotary and Radial Cylinder Engines—Practical Hints for Trouble Shooters—Engine Stoppage Analyzed—Troubles in Ignition System—Defects in Fuel System—Early Duplex Zenith O. D. Carburetor—Faults in Oiling Systems—Defects in Water-Cooling Systems—Causes of Noisy Operation—Summary of Hints for Starting Engine—Tables of Engine Troubles—Lost Power and Overheating—Noisy Operation of Powerplant—"Skipping" or Irregular Operation—Ignition System Troubles Summarized—Motor Stops Without Warning—Motor Misfires—Electrical System Components—Sparkplugs—Magneto—Storage Battery—Timer—Induction Coil—Wiring—Carburetion System Faults Summarized—Motor Stops in Flight—Motor Races—Motor Misfires—Noisy Operation.

The proper installation of the airplane powerplant is more important than is generally supposed, as while these engines are usually well balanced and run with little vibration, it is necessary that they be securely anchored and that various connections to the auxiliary parts be carefully made in order to prevent breakage from vibration and that attendant risk of motor stoppage while in the air. The type of motor to be installed determines the method of installation to be followed. As a general rule six-cylinder vertical engine and eight-cylinder Vee type are mounted in substantially the same way. The radial, fixed cylinder forms and the radial, rotary cylinder Gnome and Rhone rotary types require an entirely different method of mounting.

Early Inverted Engine Mounting.—Some unconventional mountings have been devised, notably that shown at Fig. 424, which is a six-cylinder German engine that was installed in just the opposite way to that commonly followed. The inverted cylinder construction was not generally followed in early engines because even with pressure feed, dry crankcase type lubricating system there was considerable danger of over-lubrication and of oil collecting and carbonizing in the combustion-chamber and gumming up the valve action much quicker than would be the case if the engine was operated in the conventional upright position. The reason for mounting an engine in this way is to obtain a lower center of gravity and also to make for more perfect streamlining of the front end of the fuselage in some cases. It is now used to some extent and is presented to show one of the possible systems of installing an airplane engine.

Conventional Installation.—In a number of airplanes of the tractor-biplane type the powerplant installation is not very much different than that which is found in automobile practice. The illustration at Fig. 425 is a very clear representation of the method of mounting the Curtiss eight-cylinder 90 horsepower or model OX2 engine in the fuselage of the Curtiss JN4 tractor biplane which was so generally used in the United States as

a training machine and for a number of years after the war for civilian flying. It will be observed that the fuel tank is mounted under a cowl directly behind the motor and that it feeds the carburetor by means of a flexible fuel pipe. As the tank is mounted higher than the carburetor, it will feed that member by gravity. The radiator is mounted at the front end of the fuselage and connected to the water piping on the motor by the usual rubber hose connections. An oil pan is placed under the engine and

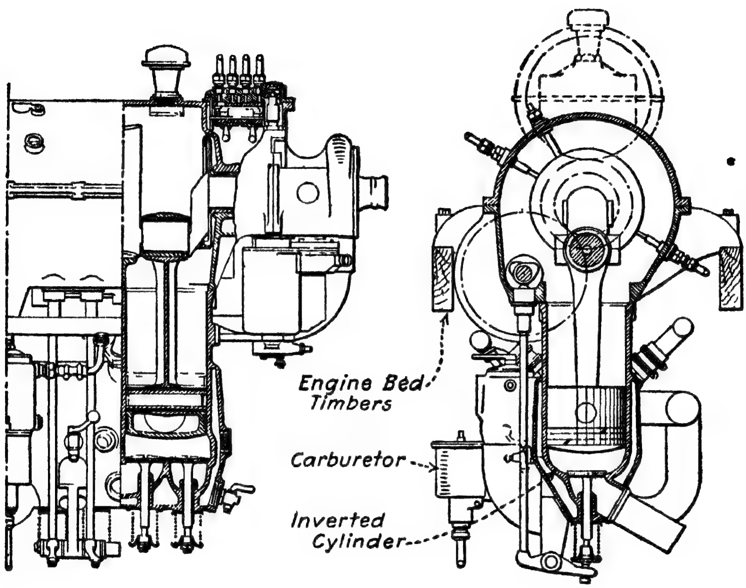


Fig. 424.—Unconventional Mounting of Early German Inverted Cylinder Motor, a Pioneer Form.

the top is covered with a hood just as in motor car practice. The panels of aluminum are attached to the sides of the fuselage and are supplied with doors which open and provide access to the carburetor, oil-gauge and other parts of the motor requiring inspection. The complete installation with the powerplant enclosed is given at Fig. 426, and in this it will be observed that the exhaust pipes are connected to discharge members that lead the gases above the top plane. In the engine shown at Fig. 425 the exhaust flows directly into the air at the sides of the machine through short pipes bolted to the exhaust gas outlet ports. The installation of the radiator just back of the tractor screw insures that adequate cooling will be obtained because of the rapid air flow due to the propeller slipstream.

Mounting Curtiss OX Series Engines.—The following instructions are given in the Curtiss Instruction Book for installing the OX series engines and preparing them for flights, and taken in connection with the very clear illustration presented no difficulty should be experienced in understanding the proper installations, and mounting of this powerplant. The bearers or beds should be two inches wide by three inches deep, preferably of lamin-

ated hard wood, and placed 11⅝ inches apart. They must be well braced. The six arms of the base of the motor are drilled for ⅜-inch bolts, and none but this size should be used.

1. *Anchoring the Motor.* Put the bolts in from the bottom, with a large washer under the head of each so the head cannot cut into the wood. On every bolt use a castellated nut and a cotter pin, or an ordinary nut and a lock washer, so the bolt will not work loose. Always set motor in place and fasten before attaching any auxiliary apparatus, such as carburetor, etc.

2. *Inspecting the Ignition-Switch Wires.* The wires leading from the ignition switch must be properly connected—one end to the motor body for ground, and the other end to the post on the breaker box of the magneto.

3. *Filling the Radiator.* Be sure that the water from the radiator fills the cylinder jackets. Pockets of air may remain in the cylinder jackets even

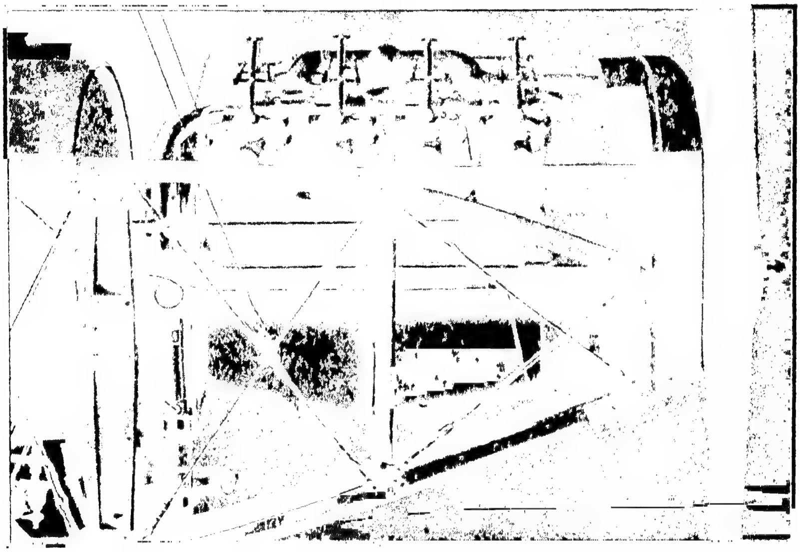


Fig. 425.—How Curtiss Model OX2 Motor was Installed in Fuselage of Curtiss Tractor Training Biplane. Note Similarity of Mounting in Respect to Radiator to Automobile Practice.

though the radiator may appear full. Turn the motor over a few times by hand after filling the radiator, and then add more water if the radiator will take it. The air pockets, if allowed to remain, may cause overheating and develop serious trouble when the motor is running.

4. *Filling the Oil Reservoir.* Oil is admitted into the crankcase through the breather tube at the rear. It is well to strain all oil put into the crankcase. In filling the oil reservoir be sure to turn the handle on the oil sight-gauge till it is at right angles with the gauge. The oil sight-gauge is on the side of the lower half of the crankcase. Put in about three gallons of the best obtainable oil, Mobile B recommended. It is important to remember that the very best oil is none too good.

5. *Oiling Exposed Moving Parts.* Oil all rocker-arm bearings before each flight. A little oil should be applied where the push rods pass through the stirrup straps.

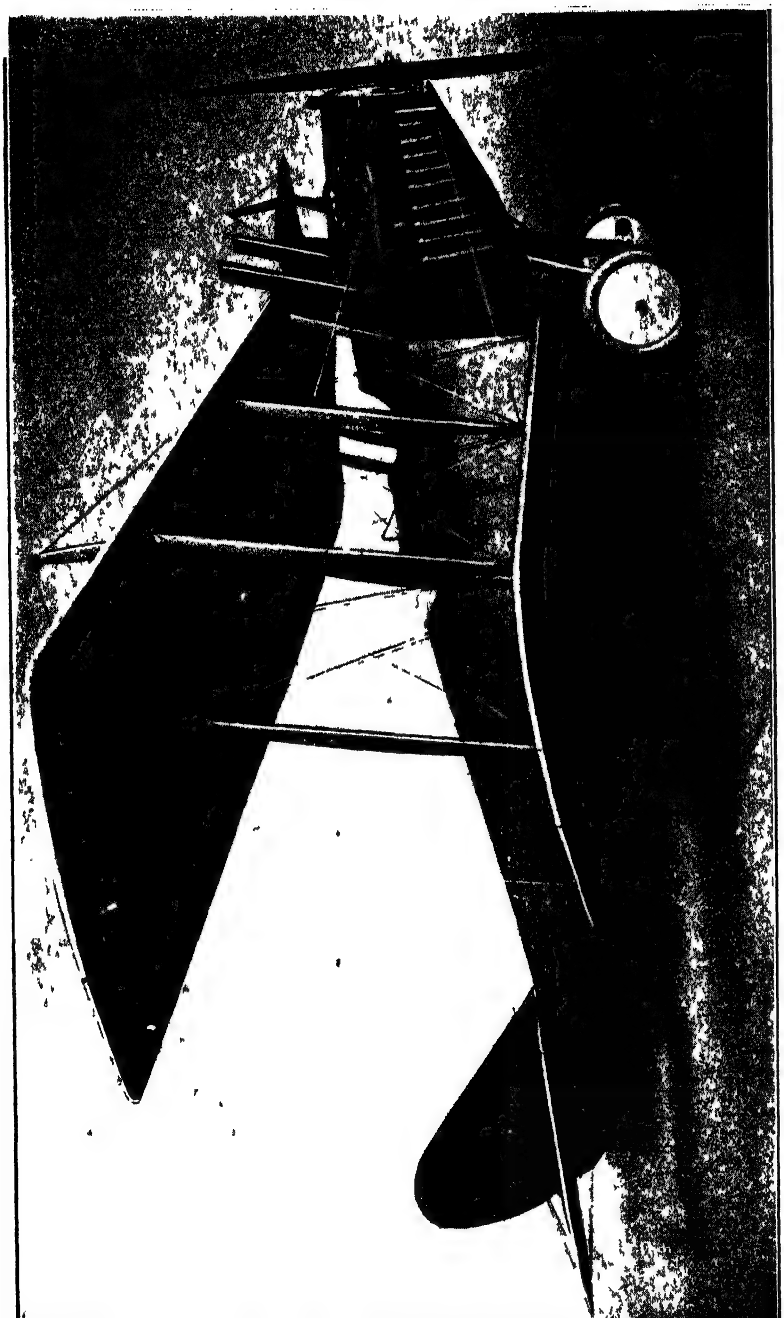


Fig. 426.—War-Time Model of Curtiss JN4 Training Machine Showing Thorough Enclosure of Powerplant in the Front End of the Fuselage, and Method of Disposing of the Exhaust Gases.

6. *Filling the Gasoline Tanks.* Be certain that all connections in the gasoline system are tight.

7. *Turning on the Gasoline.* Open the cock leading from the gasoline tank to the carburetor.

8. *Charging the Cylinders.* With the ignition switch OFF, prime the motor by squirting a little gasoline in each exhaust port and then turn the propeller backward two revolutions. Never open the exhaust valve by operating the rocker arm by hand, as the push rod is liable to come out of its socket in the cam follower and bend the rocker arm when the motor turns over.

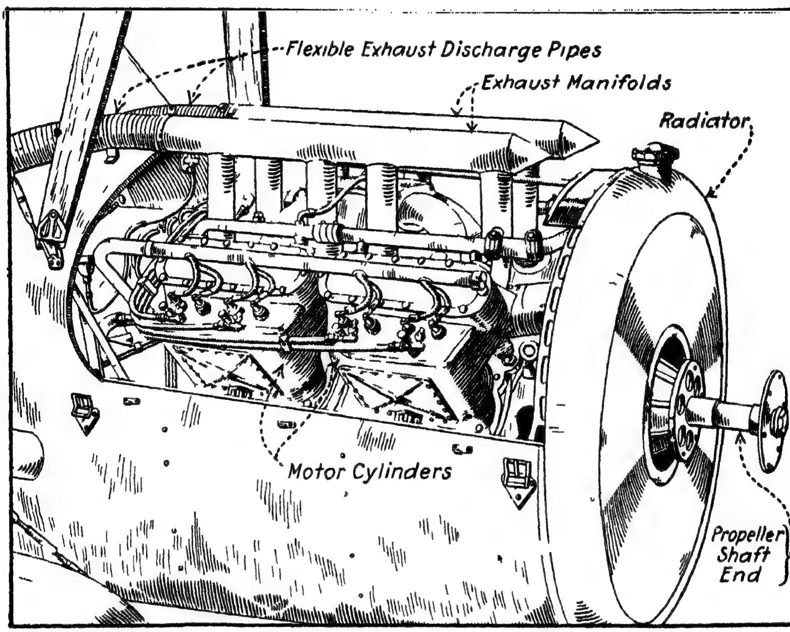


Fig. 427.—Front View of the Early LWF Tractor Biplane Fuselage, which Used Eight-Cylinder "Vee" Engine. Note Method of Disposing of Exhaust Gases.

9. *Starting the Motor by Hand.* Always retard the spark part way, to prevent back-firing, by pulling forward the wire attached to the breaker box. Failure to so retard the spark in starting may result in serious injury to the operator. Turn on the ignition switch with throttle partly open; give a quick, strong pull down and outward on the starting crank or propeller. As soon as the motor is started advance the spark by releasing the retard wire.

10. *Oil Circulation.* Let the motor run at low speed for a few minutes in order to establish oil circulation in all bearings. With all parts functioning properly, the throttle may be opened gradually for warming up before flight.

Engine Bed Dimensions.—The Society of Automotive Engineers made efforts to standardize dimensions of bed timbers for supporting powerplant in an airplane. Owing to the great difference in length no standardization

was thought possible in this regard. The dimensions recommended were as follows:

Distance between timbers	12 in.	14 in.	16 in.
Width of bed timbers	1½ in.	1¾ in.	2 in.
Distance between centers of bolts	13½ in.	15¾ in.	18 in.

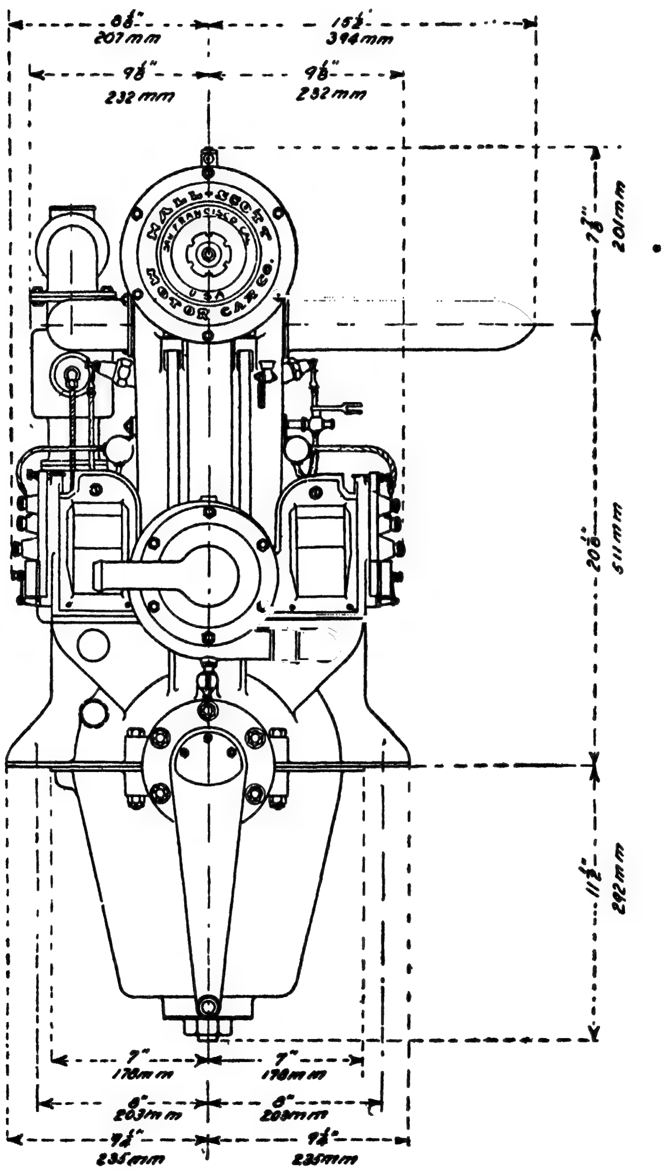


Fig. 428.—End Elevation of Hall-Scott A7 Four-Cylinder Motor Giving Principal Installation Dimensions.

It will be evident that if any standard of this nature were adopted by engine builders that the designers of fuselages could easily arrange their bed timbers to conform to these dimensions, whereas it would be difficult to have them adhere to any standard longitudinal dimensions which are much more easily varied in fuselages than the transverse dimensions are. It, however, should be possible to standardize the longitudinal positions of the holding down bolts as the engine designer would still be able to allow himself considerable space fore-and-aft of the bolts.

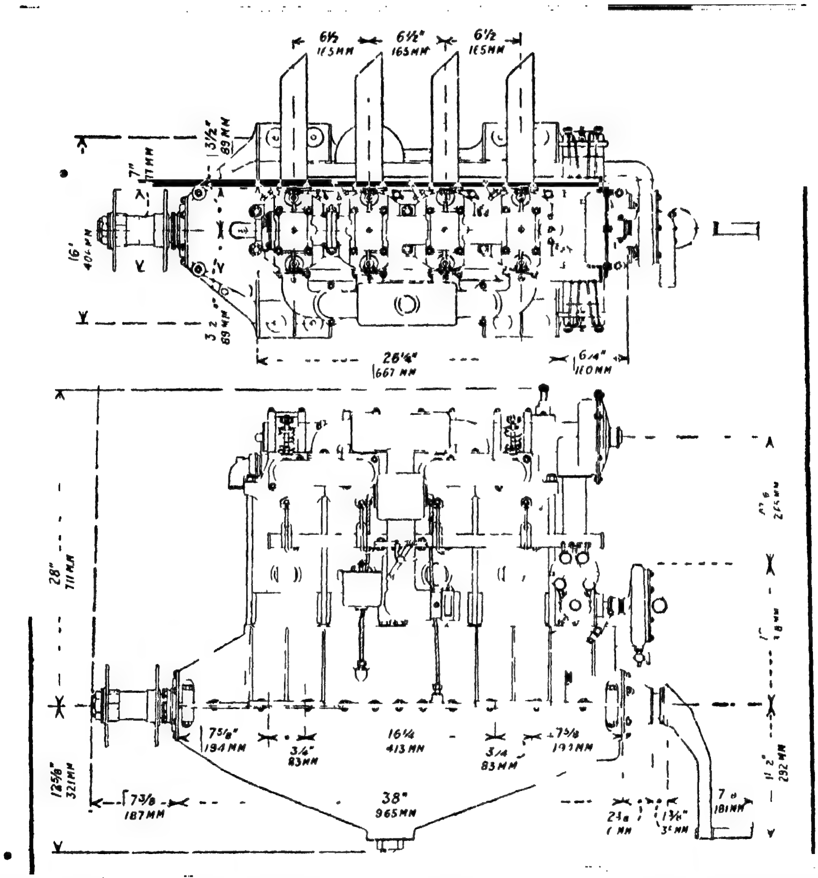


Fig. 429.—Plan and Side Elevation of Hall-Scott A7 Four-Cylinder Airplane Engine with Installation Dimensions.

Hall-Scott Engine Installation.—The very thorough manner in which the installation diagrams are prepared by the leading engine makers leaves nothing to the imagination. The dimensions of the Hall-Scott four-cylinder airplane engine are given clearly in our inch measurements with the metric equivalents at Figs. 428 and 429, the former showing a vertical elevation while the latter has a plan view and side elevation.

The dimensions of the six-cylinder Hall-Scott motor which is known as the type A5 125 horsepower are given at Fig. 430, which is a plan view.

The dimensions are given both in inch sizes and the metric equivalents. A diagram showing the location of the engine and the various pipes leading to the auxiliary groups is outlined at Fig. 431. The following instructions for installing the Hall-Scott powerplant are reproduced from the instruction book issued by the maker. Operating instructions which are given should enable any good mechanic to make a proper installation and to keep the engine in good running condition.

Fuel Supply System.—Gasoline giving the best results with this equipment is as follows: Gravity 58-62 degrees Baume A. Initial boiling point—Richmond method—102 degrees Fahrenheit. Sulphur .014. Calorimetric bomb test 20,610 B.t.u. per pound. If the gasoline tank is placed in the

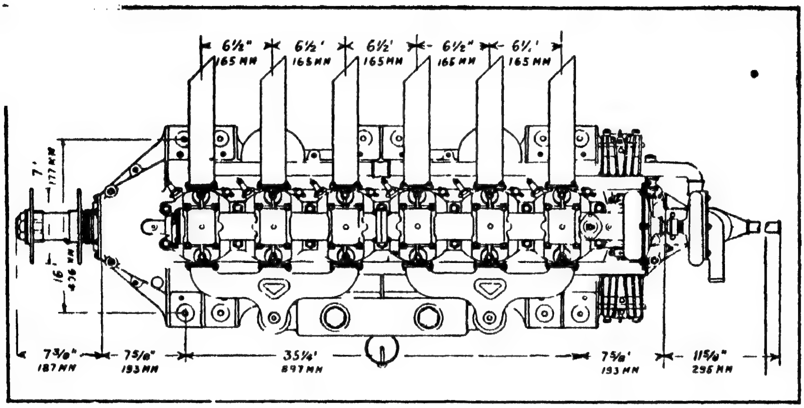


Fig. 430.—Plan View of Hall-Scott Type A5, 125 Horsepower Six-Cylinder Aviation Engine Showing Installation Dimensions.

fuselage below the level of the carburetor, a hand pump must be used to maintain air pressure in gas tank to force the gasoline to the carburetor. After starting the engine the small auxiliary air pump upon the engine will maintain sufficient pressure. A7a and A5a engines are furnished with a new type auxiliary air pump. This should be frequently oiled and care taken so no grit or sand will enter which might lodge between the valve and its seat, which would make it fail to operate properly. An air relief valve is furnished with each engine. It should be screwed into the gas tank and properly regulated to maintain the pressure required. This is done by screwing the ratchet on top either up or down. If two tanks are used in a plane one should be installed in each tank. All air pump lines should be carefully gone over quite frequently to ascertain if they are tight. Check valves have to be placed in these lines. In some cases the gasoline tank is placed above the engine, allowing it to drain by gravity to the carburetor. When using this system there should be a drop of not less than two feet from the lowest portion of the gasoline tank to the upper part of the carburetor float chamber. Even this height might not be sufficient to maintain the proper volume of gasoline to the carburetor at high speeds. Air pressure is advised upon all tanks to insure the proper supply of gasoline. When using gravity feed without air pressure be sure to vent the tank to allow circulation of air. If gravity tank is used and the engine runs satis-

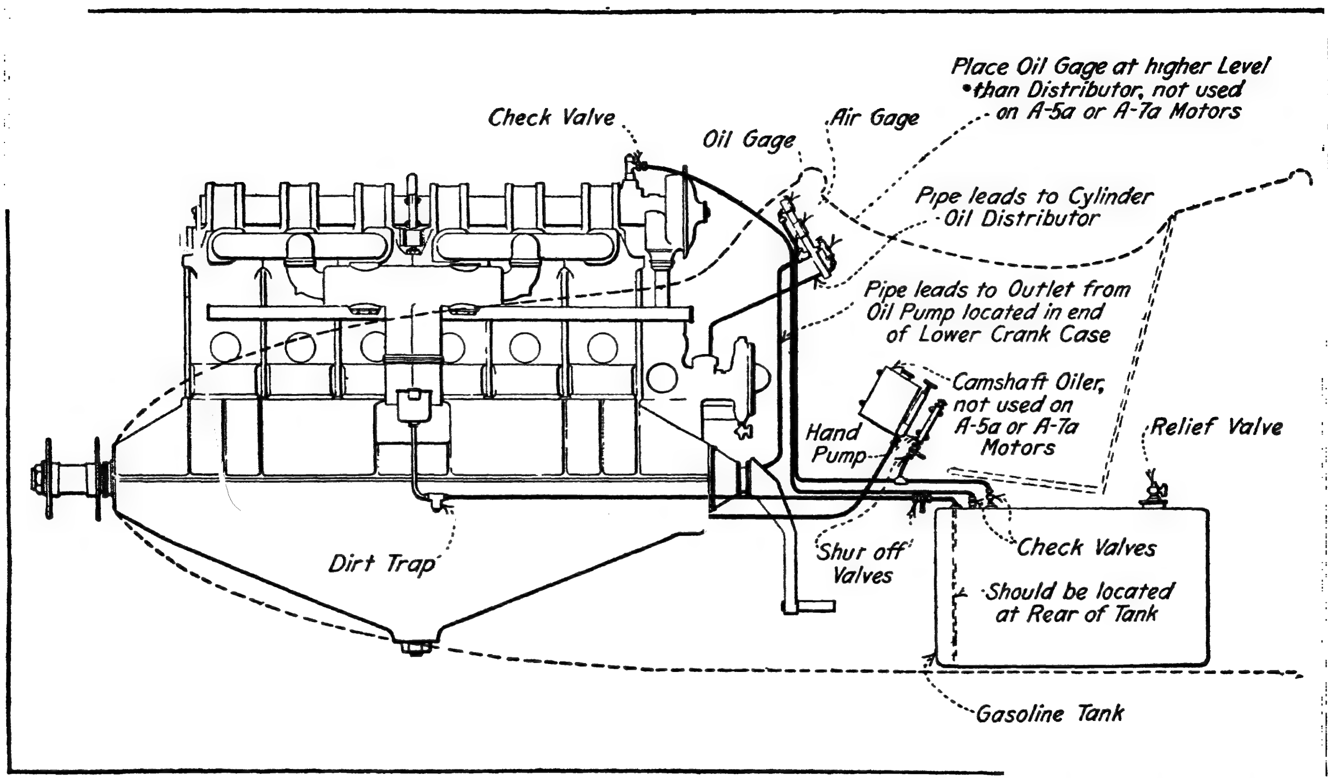


Fig. 431.—Diagram Showing Hall-Scott Type A5, 125 Horsepower Engine with Pressure Feed Fuel Supply System, an Early Installation.

factorily at low speeds but cuts out at high speeds the trouble is undoubtedly due to insufficient height of the tank above the carburetor. The tank should be raised or air pressure system used.

Ignition Switches.—Two "Dixie" switches are furnished with each engine. Both of these should be installed in the pilot's seat, one controlling the R. H., and the other the L. H. magneto. By shorting either one or the other it can be quickly determined if both magnetos, with their respective sparkplugs, are working correctly. Care should be taken not to use sparkplugs having *special extensions or long protruding points.* Plugs giving best results are extremely small with short points.

Hall-Scott Water System.—A temperature gauge should be installed in the water pipe, coming directly from the cylinder nearest the propeller. Installing this instrument in the radiator cap has not always given satisfactory results. This is especially noticeable when the water in the radiator becomes low, not allowing it to touch the bulb on the moto-meter. For ordinary running, it should not indicate over 150 degrees Fahrenheit. In climbing tests, however, a temperature of 160 degrees Fahrenheit can be maintained without any ill effects upon the engine. In case the engine becomes overheated, the indicator will register above 180 degrees Fahrenheit, in which case it should be stopped immediately. Overheating is most generally caused by retarded spark, excessive carbon in the cylinders, insufficient lubrication, improperly timed valves, lack of water, clogging of water system in any way which would obstruct the free circulation of the water.

Overheating will cause the engine to knock, with possible damaging results. Suction pipes should be made out of thin tubing, and run within a quarter or an eighth of an inch of each other, so that when a hose is placed over the two, it will not be possible to suck together. This is often the case when a long rubber hose is used, which causes overheating. Radiators should be flushed out and cleaned thoroughly quite often. A dirty radiator may cause overheating.

When filling the radiator it is very important to remove the plug on top of the water pump until water appears. This is to avoid air pockets being formed in the circulating system, which might not only heat up the engine, but cause considerable damage. All water pump hoses and connections should be tightly taped and shellacked after the engine is properly installed in the plane. The greatest care should be taken when making engine installation *not* to use smaller inside diameter hose connection than water pump suction end casting. One inch and a quarter inside diameter should be used on A7 and A5 motors, while nothing less than one inch and a half inside diameter hose or tubing on all A7a and A5a engines. It is further important to have light spun tubing, void of any sharp turns, leads from pump to radiator and cylinder water outlet to radiator. In other words, the water circulation through the engine must be as little restricted as possible. Be sure no light hose is used, that will often suck together when engine is started. To thoroughly drain the water from the entire system, open the drain cock at the lowest side of the water pump.

Preparations to Start Engine.—Always replenish gasoline tanks through a strainer which is clean. This strainer must catch all water and other

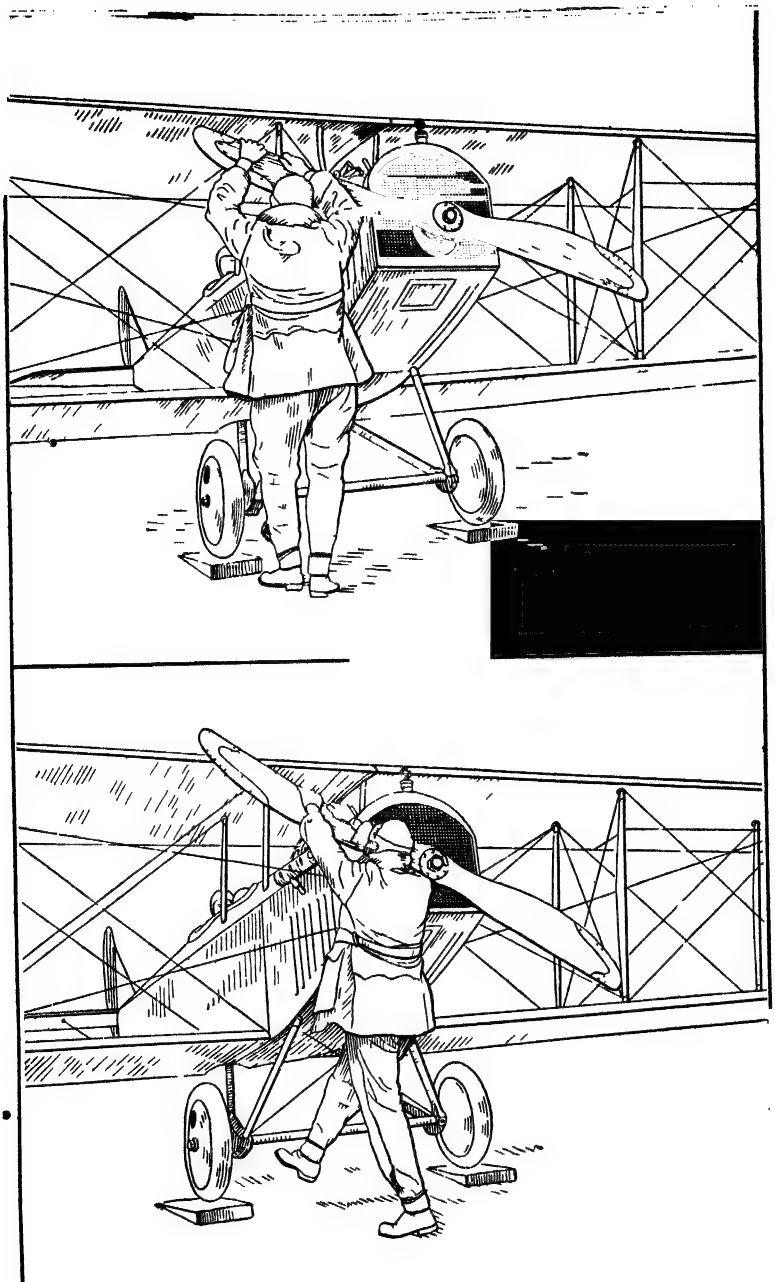


Fig. 432.—Illustration Depicting Wrong and Right Methods of "Swinging the Stick" to Start an Airplane Engine. At Top, Poor Position to Get Full Throw and Get Out of the Way. Below, Correct Position to Get Quick Turn Over of the Crankshaft and Spring Back from Rotating Propeller.

impurities in the gasoline. Pour at least three gallons of fresh oil into the lower crankcase. Oil all rocker arms through oilers upon rocker arm housing caps. Be sure radiators are filled within one inch of the top. After all the parts are oiled, and the tanks filled, the following must be looked after before starting: See if crankshaft flange is tight on shaft. See if propeller bolts are tight and evenly drawn up. See if propeller bolts are wired. See if propeller is trued up to within ⅛ inch.

Every four days the magnetos should be oiled if the engine is in daily use. Every month all cylinder hold-down nuts should be gone over to ascertain if they are tight. (Be sure to recotter nuts.) See if magnetos are bolted on tight and wired. See if magneto cables are in good condition. See if rocker arm tappets have a .020 inch clearance from valve stem when valve is seated. See if tappet clamp screws are tight and cottered. See if all gasoline, oil, water pipes and connections are in perfect condition. Air on gas line should be tested for leaks. Pump at least three pounds air pressure into gasoline tank. After making sure that above rules have been observed, test compression of cylinders by turning propeller.

Be sure all compression release and priming cocks do not leak compression. If they do, replace same with a new one immediately, as this might cause premature firing. Open priming cocks and squirt some gasoline into each. Close cocks. Open compression release cocks. Open throttle slightly. If using Berling magnetos they should be three-quarters advanced. If all the foregoing directions have been carefully followed, the engine is ready for starting. In cranking engine either by starting crank, or propeller, it is essential to throw it over compression quickly. Immediately upon starting, close compression release cocks. When engine is running, advance magnetos.

After it has warmed up, short one magneto and then the other, to be sure both magnetos and sparkplugs are firing properly. If there is a miss, the fouled plug must be located and cleaned. There is a possibility that the jets in the carburetor are stopped up. If this is the case, do not attempt to clean same with any sharp instrument. If this is done, it might change the opening in the jets, thus spoiling the adjustment. Jets and nozzles should be blown out with air or steam. An open intake or exhaust valve, which might have become sluggish or stuck from carbon, might cause trouble. Be sure to remedy this at once by using a little coal-oil or kerosene on same, working the valve by hand until it becomes free. We recommend using graphite mixed with oil on valve stems to guard against sticking or undue wear.

Installing Early Rotary and Radial Cylinder Engines.—When rotary engines are installed simple steel stamping or "spiders" are attached to the fuselage to hold the fixed crankshaft. Inasmuch as the motor projects clear of the fuselage proper there is plenty of room back of the front spider plate to install the auxiliary parts such as the oil pump, air pump and ignition magneto and also the fuel and oil containers. The diagram given at Fig. 433 shows how a Gnome "monosoupape" engine is installed on the anchorage plates and it also outlines clearly the piping necessary to convey the oil and fuel and also the air-piping needed to put pressure on both fuel and

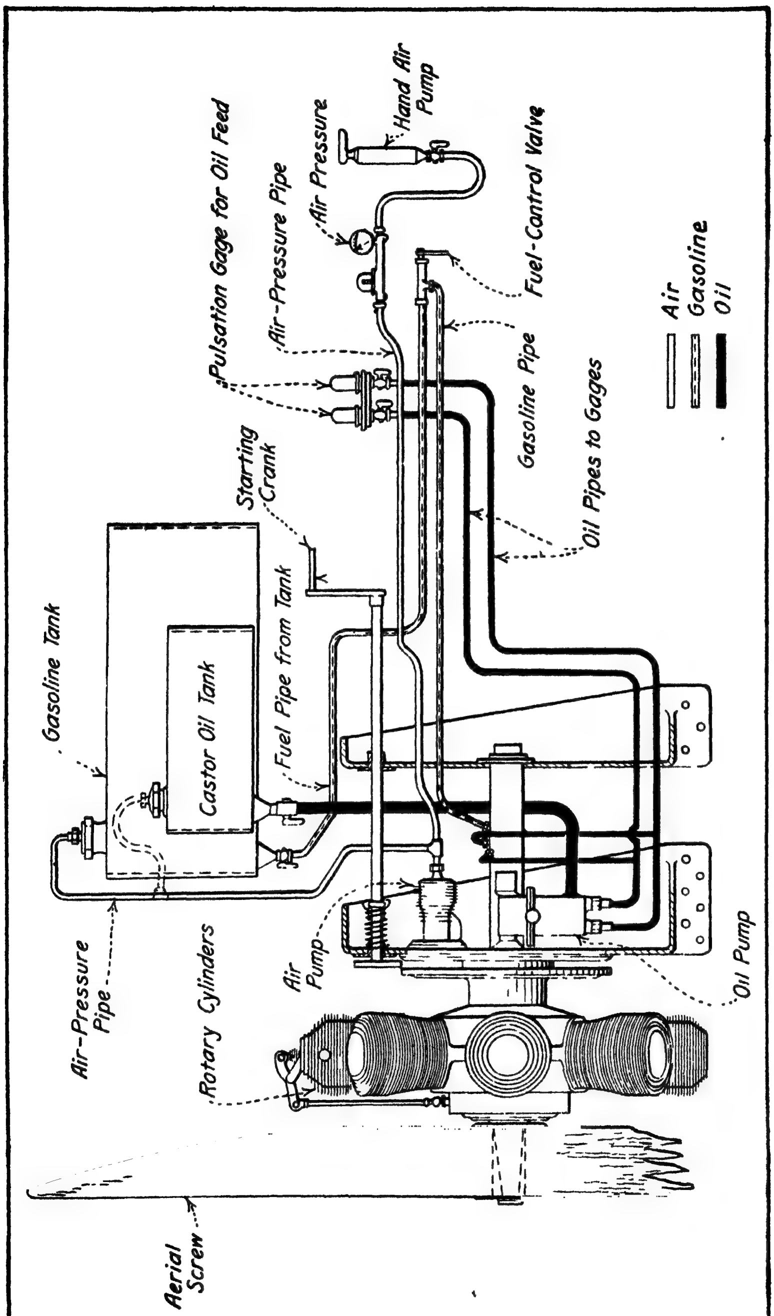


Fig. 433.—Diagram Defining Method of Installing Early Gnome "Monosoupape" Motor in Tractor Biplane. Note Necessary Pipes for Fuel, Oil and Air Lines, Also Location of Fuel and Oil Tanks.

oil tanks to insure positive supply of these liquids which may be carried in tanks placed lower than the motor in some installations.

The diagrams given at Figs. 434 and 435 show other mountings of Gnome engines and are self-explanatory. The simple mounting possible when the Anzani ten-cylinder radial fixed type engine is used is given at Fig. 436. The front end of the fuselage is provided with a substantial pressed steel plate having members projecting from it which may be bolted to the longerons. The bolts that hold the two halves of the crankcase together project through the steel plate and hold the engine securely to the front end of the fuselage.

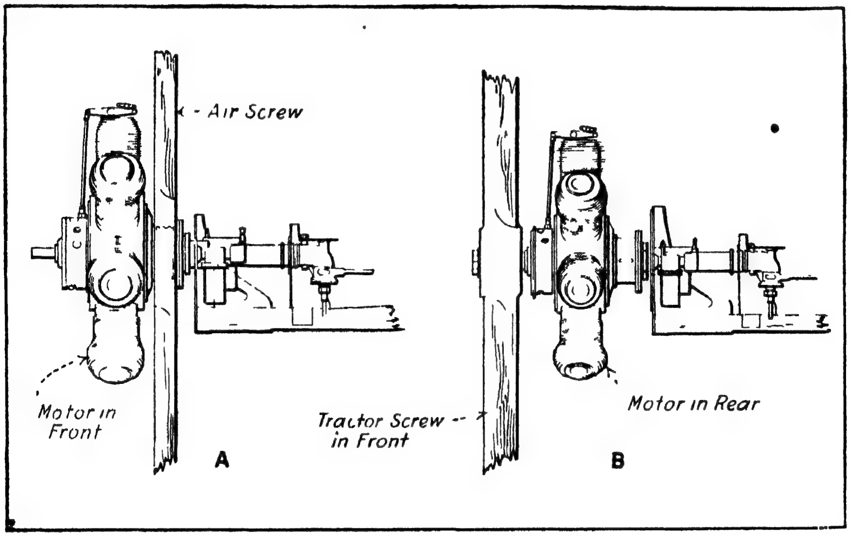


Fig. 434.—Diagram Showing Two Methods of Placing Propeller on Gnome Rotary Motor. A—Motor in Front of Air Screw. B—Common Method in Which Tractor Screw was Placed in Front of the Motor.

Practical Hints for Trouble Shooters.—One who is not thoroughly familiar with engine construction will seldom locate troubles by haphazard experimenting and it is only by a systematic search that the cause can be discovered and the defects eliminated. In this chapter the writer proposes to outline some of the most common powerplant troubles and to give sufficient advice to enable those who are not thoroughly informed to locate them by a logical process of elimination. The internal-combustion motor, which is the powerplant of all gasoline automobiles as well as airplanes, is composed of a number of distinct groups, which in turn include distinct components. These various appliances are so closely related to each other that defective action of any one may interrupt the operation of the entire powerplant. Some of the auxiliary groups are more necessary than others and the powerplant will continue to operate for a time even after the failure of some important parts of some of the auxiliary groups. The gasoline-engine in itself is a complete mechanism, but it is evident that it cannot deliver any power without some means of supplying gas to the cylinders and igniting the compressed gas charge after it has been compressed in the cylinders. From

this it is patent than the ignition and carburetion systems are just as essential parts of the powerplant as the piston, connecting rod, or cylinder of the motor. The failure of either the carburetor or igniting means to function properly will be immediately apparent by faulty action of the powerplant.

To insure that the motor will continue to operate it is necessary to keep it from overheating by some form of cooling system and to supply oil to the moving parts to reduce friction. The cooling and lubrication groups are not so important as carburetion and ignition, as the engine would run for a limited period of time even should the cooling system fail or the oil supply cease. It would only be a few moments, however, before the engine would overheat if the cooling system was at fault, and the parts seize if the lubricating system should fail. Any derangement in the carburetor or

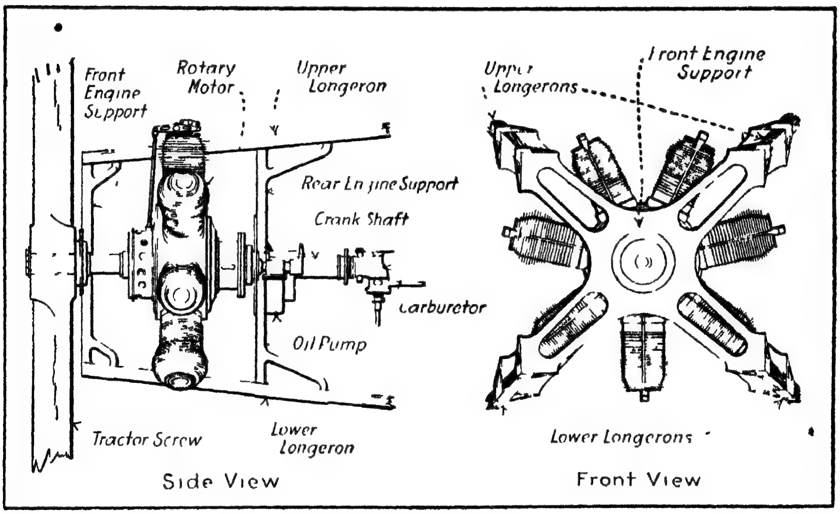


Fig. 435.—How Gnome Rotary Motor was Sometimes Attached to Airplane Fuselage, so Complete Enclosure Could be Secured by Cowling.

ignition mechanism would manifest itself at once because the engine operation would be affected, but a defect in the cooling or oiling system would not be noticed so readily. The careful aviator will always inspect the motor mechanism before starting on a trip of any consequence, and if inspection is carefully carried out and loose parts tightened it is seldom that irregular operation will be found due to actual breakage of any of the components of the mechanism. Deterioration due to natural causes matures slowly, and sufficient warning is always given when parts begin to wear so satisfactory repairs may be promptly made before serious derangement or failure is manifested.

Engine Stoppage Analyzed.—Before describing the points that may fail in the various auxiliary systems it will be well to assume a typical case of engine failure and show the process of locating the trouble in a systematic manner by indicating the various steps which are in logical order and which could reasonably be followed. In any case of engine failure the ignition

system, motor compression, and carburetor should be tested first. If the ignition system is functioning properly one should determine the amount of compression in all cylinders and if this is satisfactory the carbureting group should be tested. If the ignition system is working properly and there is a decided resistance in the cylinders when the propeller is turned, proving that there is good compression, one may suspect the carburetor.

If the carburetor appears to be in good condition, the trouble may be caused by the ignition being out of time, which condition is possible when the magneto timing gear or coupling is attached to the armature shaft by a taper and nut retention instead of the more positive key or taper-pin fastening. It is possible that the inlet manifold may be broken or perforated, that an exhaust valve is stuck on its seat because of a broken or bent stem, broken or loose cam, or failure of the camshaft drive because the teeth are stripped from the engine shaft or camshaft gears; or because the key or other fastening on either gear has failed, allowing that member to turn independently of the shaft to which it normally is attached. The gasoline feed pipe may be clogged or broken, the fuel supply may be depleted, or the shut-off cock in the gasoline line may have jarred closed. The gasoline filter may be filled with dirt or water which prevents passage of the fuel.

The defects outlined above, except the failure of the gasoline supply, are very rare, and if the container is found to contain fuel and the pipe line to be clear to the carburetor, it is safe to assume the vaporizing device is at fault. If fuel continually runs out of the mixing chamber the carburetor is said to be flooded. This condition results from failure of the shut-off needle to seat properly or from a punctured hollow metal float. It is possible that not enough gasoline is present in the float chamber. If the passage controlled by the float-needle valve is clogged or if the float was badly out of adjustment, this contingency would be probable. When the carburetor is examined, if the gasoline level appears to be at the proper height, one may suspect that a particle of lint, or dust, or fine scale, or rust from the gasoline tank has clogged the bore of the main jet in the mixing chamber.

If the ignition system and carburetor appear to be in good working order, and hand cranking shows that there is no compression in one or more of the cylinders, it means some defect in the valve system. If the engine is a multiple-cylinder type and one finds poor compression in all of the cylinders it may be due to the rare defect of improper valve timing. This may be caused by a gear having altered its position on the camshaft or crankshaft, because of a sheared key or pin having permitted the gear to turn about half of a revolution and then having caught and held the gear in place by a broken or jagged end so that camshaft would turn, but the valves open at the wrong time.

If but one of the cylinders is at fault and the rest appear to have good compression the trouble may be due to a defective condition either inside or outside of that cylinder. The external parts may be inspected easily, so the following should be looked for; a broken valve, a warped valve-head, broken valve-springs, sticking or bent valve-stems, dirt under valve-seat, leak at valve-chamber cap or sparkplug gasket. Defective priming cock, cracked cylinder head (rarely occurs), leak through cracked sparkplug in-

sulation, valve-plunger stuck in the guide, lack of clearance between valve-stem end and top of plunger caused by loose adjusting screw which has worked out of place and kept the valve from seating. The faulty compression may be due to defects inside the motor. The piston-head may be cracked (rarely occurs), piston rings may be broken, the slots in the piston rings may be in line, the rings may have lost their elasticity or have become gummed in the grooves of the piston, or the piston and cylinder walls may be badly scored by a loose wristpin or by defective lubrication. If the

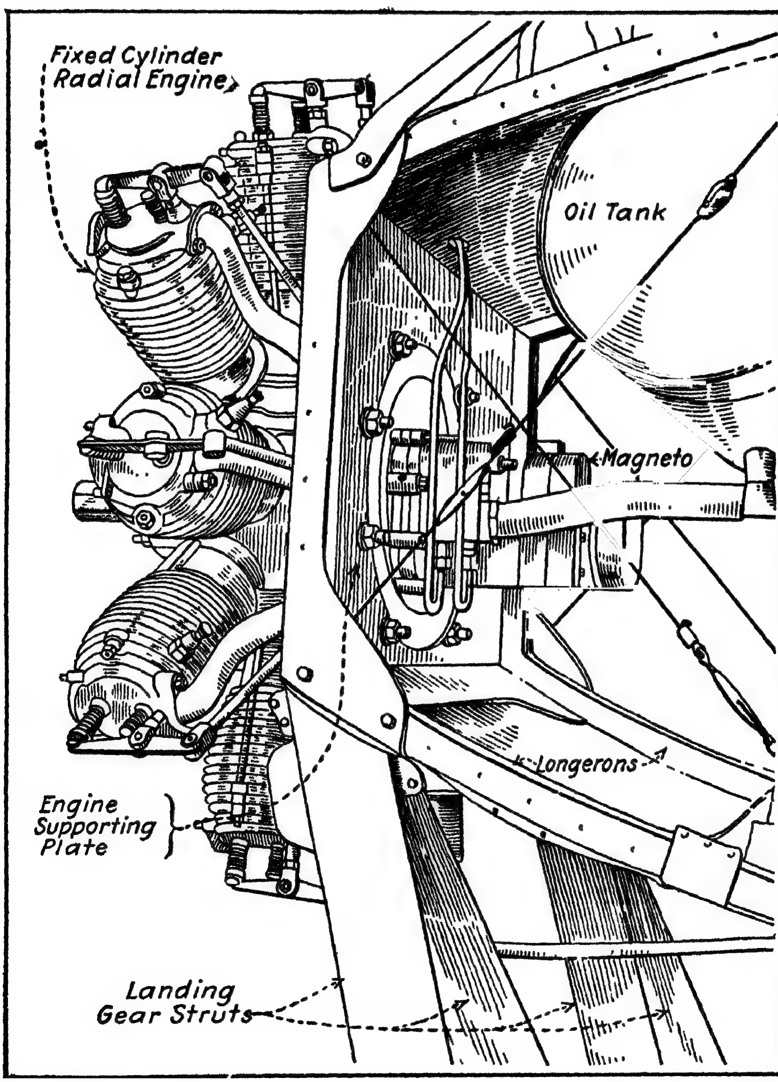


Fig. 436.—How the Early Anzani Ten-Cylinder Radial Engine was Fastened to a Steel Plate Securely Attached to the Longerons in the Front End of Tractor Airplane Fuselage.

motor is a type with a separate head it is possible the gasket or packing between the cylinder and combustion-chamber may leak, either admitting water to the cylinder or allowing compression to escape. This construction is not often found in modern airplane motors.

Troubles in Ignition System.—If the first test of the motor had showed that the compression was as it should be and that there were no serious mechanical defects and there was plenty of gasoline at the carburetor, this would have demonstrated that the ignition system was not functioning properly. If a battery is employed to supply current the first step is to take the sparkplugs out of the cylinders and test the system by turning over the engine by hand. If there is no spark in any of the plugs, this may be considered a positive indication that there is a broken main current lead from the battery, a defective ground connection, a loose battery terminal, or a broken connector. If none of these conditions are present, it is safe to say that the battery is no longer capable of delivering current. While magneto ignition is generally used on airplane engines, there is apt to be some development of battery ignition, especially on engines equipped with electric self-starters which are now being experimented with.

The sparkplugs may be short circuited by cracked insulation or carbon and oil deposits around the electrode. The secondary wires may be broken or have defective insulation which permits the current to ground to some metal part of the fuselage or motor. The electrodes of the sparkplug may be too far apart to permit a spark to overcome the resistance of the compressed gas, even if a spark jumps the air space, when the plug is laid on the cylinder.

If magnetos are fitted as is usually the case at present and a spark is obtained between the points of the plug and that device or the wire leading to it from the magneto is in proper condition, the trouble is probably caused by the magneto being out of time. This may result if the driving gear is loose on the armature-shaft or crankshaft, and is a rare occurrence. If no spark is produced at the plugs the secondary wire may be broken, the ground wire may make contact with some metallic portion of the fuselage before it reaches the switch, the carbon collecting brushes may be broken or not making contact, the contact points of the make-and-break device may be out of adjustment, the wiring may be attached to wrong terminals, the distributor filled with metallic particles, carbon, dust or oil accumulations, the distributor contacts may not be making proper connection because of wear and there may be a more serious derangement, such as a burned out secondary winding or a punctured condenser.

If the motor runs intermittently, *i.e.*, starts and runs only a few revolutions, aside from the conditions previously outlined, defective operation may be due to seizing between parts because of insufficient oil or deficient cooling, too much oil in the crankcase which fouls the cylinder after the crankshaft has revolved a few turns, and derangements in the ignition or carburetion systems that may be easily remedied. There are a number of defective conditions which may exist in the ignition group, that will result in "skipping" or irregular operation and the following points should be considered first: weak source of current due to worn out dry cells or discharged storage batteries; weak magnets in magneto, or defective contacts

at magneto; dirt in magneto distributor or poor contact at collecting brushes. Dirty or cracked insulator at sparkplug will cause short circuit and can only be detected by careful examination. The following points should also be checked over when the plug is inspected: Excessive space between electrodes, points too close together, loose central electrodes, or loose point on plug body, soot or oil particles between electrodes, or on the surface of the insulator, cracked insulator, oil or water on outside of insulator. Short circuits in the condenser or internal wiring of induction coils or magnetos, which are fortunately not common, can seldom be remedied except at the factory where these devices were made. If an engine stops suddenly and the defect is in the ignition system the trouble is usually never more serious than a broken or loose wire. This may be easily located by inspecting the wiring at the terminals. Irregular operation or misfiring is harder to locate because the trouble can only be found after the many possible defective conditions have been checked over, one by one.

Defects in Fuel Systems.—Defective carburetion often causes misfiring or irregular operation. The common derangement of the components of the fuel system that are common enough to warrant suspicion and the best methods for their location follows: First, disconnect the feed pipe from the carburetor and see if the gasoline flows freely from the tank. If the stream coming out of the pipe is not the full size of the orifice it is an indication that the pipe is clogged with dirt or that there is an accumulation of rust, scale, or lint in the strainer screens of the filter. It is also possible that the fuel shut-off valve may be wholly or partly closed. If the gasoline flows by gravity the liquid may be air bound in the tank, while if a pressure-feed system is utilized the tank may leak so that it does not retain pressure; the check valve retaining the pressure may be defective or the pipe conveying the air or gas under pressure to the tank may be clogged.

If the gasoline flows from the pipe in a steady stream the carburetor demands examination. There may be dirt or water in the float chamber, which will constrict the passage between the float chamber and the spray nozzle, or a particle of foreign matter may have entered the main or a metering plug nozzle and stopped up the fine holes therein. The float may bind on its guide, the needle valve regulating the gasoline-inlet opening in bowl may stick to its seat. Any of the conditions mentioned would cut down the gasoline supply and the engine would not receive sufficient quantities of gas. The air-valve spring may be weak or the air valve broken. The gasoline-adjusting needle may be loose and jar out of adjustment, or the air-valve spring-adjusting nuts may be such a poor fit on the stem that adjustments will not be retained. These suggestions apply only to carburetors having air valves and mixture regulating means which are used only in rare instances in airplane work. Air may leak in through the manifold, due to a porous casting, or leaky joints in a built up form and dilute the mixture. Sometimes air leaks in through worn intake valve guides. The air-intake dust screen may be so clogged with dirt and lint that not enough air will pass through the mesh. Water or sediment in the gasoline will cause misfiring because the fuel feed varies when the water or dirt constricts the standpipe bore or a particle of sediment flows in and out of one of the metering plugs.

It is possible that the carburetor may be out of adjustment. If clouds of black smoke are emitted at the exhaust pipe it is positive indication that too much gasoline is being supplied the mixture and the supply should be cut down by supplying smaller nozzle on types where this method of regulation is provided, and by making sure that the fuel level is at the proper height, or that the proper metering plug is used in those forms where the spray nozzle has no means of adjustment. If the mixture contains too much air there will be a pronounced popping back in the carburetor. When a carburetor is properly adjusted and the mixture delivered the cylinder burns properly, the exhaust gas will be clean and free from the objectionable odor present when gasoline is burned in excess. The reader is referred to the chapters on carburetors preceding this one and should study the Zenith and Stromberg carburetors which are widely used in aviation engine carburetion systems. The character of combustion may be judged by the color of the flame which issues from the exhaust ports or stacks when the engine is running with an open throttle after nightfall. If the flame is red, it indicates too much gasoline. If yellowish, it shows an excess of air, while a properly proportioned mixture will be evidenced by a pronounced blue flame, such as given by a properly adjusted blow torch or gas-stove burner.

Early Duplex Zenith Carburetor.—The Duplex Model O. D. Zenith carburetor used upon most of the six- and eight-cylinder airplane engines consists of a single float chamber, and a single air intake, joined to two separate and distinct spray nozzles, venturi and idling adjustments. It is to be noted that as the carburetor barrels are arranged side by side, both valves are mounted on the same shaft, and work in unison through a single operating lever. It is not necessary to alter their position. In order to make the engine idle well, it is essential that the ignition, especially the sparkplugs, should be in good condition. The gaskets between carburetor and manifold, and between manifold and cylinders should be absolutely air-tight. The adjustment for low speed on the carburetor is made by turning in or out the two knurled screws, placed one on each side of the float chamber. After starting the engine and allowing it to become thoroughly warmed, one side of the carburetor should be adjusted so that the three or four cylinders it feeds fire properly at low speed. The other side should be adjusted in the same manner until all six or eight cylinders fire perfectly at low speed. As the adjustment is changed on the knurled screw a difference in the idling of the engine should be noticed. If the engine begins to run evenly or speeds up it shows that the mixture becomes right in its proportion.

Be sure the butterfly throttle is closed as far as possible by screwing out the stop screw which regulates the closed position for idling. Care should be taken to have the butterfly held firmly against this stop screw at all times while idling engine. If three cylinders seem to run irregularly after changing the position of the butterfly, still another adjustment may have to be made with the knurled screw. Unscrewing this makes the mixture leaner. Screwing in closes off some of the air supply to the idling jet, making it richer. After one side has been made to idle satisfactorily repeat the same procedure with the opposite three or four cylinders. In other words, each side should be idled independently to about the same speed.

Remember that the main jet and compensating jet have no appreciable effect on the idling of the engine. The idling mixture is drawn directly through the opening determined by the knurled screw and enters the carburetor barrel through the small hole at the edge of each butterfly. This is called the priming hole and is only effective during idling. Beyond that point the suction is transferred to the main jet and compensator, which controls the power of the engine beyond the idling position of the throttle.

Faults in Oiling Systems.—While troubles existing in the ignition or carburetion groups are usually denoted by imperfect operation of the motor, such as lost power, and misfiring, derangements of the lubrication or cooling systems are usually evident by overheating, diminution in engine capacity, or noisy operation. Overheating may be caused sometimes by poor carburetion as much as by deficient cooling or insufficient oiling. When the oiling group is not functioning as it should the friction between the motor parts produces heat. If the cooling system is in proper condition, as will be evidenced by the amount of water in the radiator, and the carburetion group appears to be in good condition, the overheating is probably caused by some defect in the oiling system. Overheating is evidenced by engine temperature gauges, some showing water temperature, others, in air-cooled engines, showing heat in engine oil.

The conditions that most commonly result in poor lubrication are: Insufficient or improper oil in the engine crankcase or sump, broken or clogged oil pipes, screen at filter filled with lint or dirt, broken oil pump, or defective oil-pump drive. The supply of oil may be reduced by a defective inlet or discharge-check valve at the mechanical oiler or worn pumps. A clogged oil passage or pipe leading to an important bearing point will cause trouble because the oil cannot get between the working surfaces. It is well to remember that much of the trouble caused by defective oiling may be prevented by using only the best grades of lubricant, and even if all parts of the oil system are working properly, oils of poor quality will cause friction and overheating. The oil filter screens at various points in the line should be thoroughly cleaned at periodical intervals as recommended by the engine builders.

Defects in Water-Cooling Systems.—Cooling systems are very simple and are not liable to give trouble as a rule if the radiator is kept full of clean water and the circulation is not impeded. When overheating is due to defective cooling the most common troubles are those that impede water circulation. If the radiator is clogged or the piping of water jackets filled with rust or sediment the speed of water circulation will be slow, which will also be the case if the water pump or its driving means fail. Any scale or sediment in the water jackets or in the piping or radiator passages will reduce the heat conductivity of the metal exposed to the air, and the water will not be cooled as quickly as though the scale or rust was not present.

The rubber hose commonly used in making the flexible connections demanded between the radiator and water manifolds of the engine may deteriorate inside and particles of rubber hang down that will reduce the area of the passage. The grease from the grease cups mounted on the pump-shaft bearing to lubricate that member often finds its way into the water system and rots the inner walls of the rubber hose, this resulting in strips of the

partly decomposed rubber lining hanging down and restricting the passage. The cooling system is prone to overheat after antifreezing solutions of which calcium chloride forms a part have been used. This is due to the formation of crystals of salt in the radiator passages or water jackets, and these crystals can only be dissolved by suitable chemical means, or removed by scraping when the construction permits. Alcohol-glycerine antifreezing solutions are best for aviation engines.

Overheating is often caused by some condition in the fuel system that produces too rich or too lean mixture. Excess gasoline may be supplied if any of the following conditions are present: Bore of spray nozzle or standpipe too large, gasoline level too high, loose regulating valve, punctured sheet-metal float, dirt under float control shut-off valve or insufficient air supply because of a small venturi. If pressure feed is utilized there may be too much pressure in the tank, or the float controlled mechanism operating the shut-off in the float bowl of the carburetor may not act quickly enough.

Causes of Noisy Operation.—There are a number of powerplant derangements which give positive indication because of noisy operation. Any knocking or rattling sounds are usually produced by wear in connecting rods or main bearings of the engine, though sometimes a sharp metallic knock, which is very much the same as that produced by a loose bearing, is due to carbon deposits in the cylinder heads, or premature ignition due to advanced spark time. Squeaking sounds invariably indicate dry bearings, and whenever such a sound is heard it should be immediately located and oil applied to the parts thus denoting their dry condition. Whistling or blowing sounds are produced by leaks, either in the engine itself or in the gas manifolds. A sharp whistle denotes the escape of gas under pressure and is usually caused by a defective packing or gasket that seals a portion of the combustion-chamber or that is used for a joint at the exhaust manifold. A blowing sound indicates a leaky packing in crankcase. Grinding noises in the motor are usually caused by the timing gears and will obtain if these gears are dry or if they have become worn. Whenever a loud knocking sound is heard careful inspection should be made to locate the cause of the trouble. Much harm may be done in a few minutes if the engine is run with loose connecting rod or bearings that would be prevented by taking up the wear or looseness between the parts by the means of adjustment provided.

Summary of Hints for Starting Engine.—First make sure that all cylinders have compression. To ascertain this, open relief cocks of all cylinders except the one to be tested, crank over motor and see that a strong opposition to cranking is met with once in two revolutions. If motor has no pet cocks, crank and notice that oppositions are met at equal distances, two to every revolution of the propeller in a four-cylinder motor, three in a six-cylinder and four in an eight-cylinder. If compression is lacking, examine the parts of the cylinder or cylinders at fault in the following order, trying to start the motor whenever any one fault is found and remedied. See that the valve push rods or rocker arms do not touch valve stems for more than approximately one-half revolution in every two revolutions, and that there is not more than .010 to .020 inch clearance between them depending on the

make of the motor. Make sure that the exhaust valves seat. To determine this examine the spring or springs and see that connection to the valve stem is made properly. Take out valve and see that there is no obstruction, such as carbon, on its seat. See that valve works freely in its guide. Examine inlet valve in same manner. Listen for hissing sound while cranking motor for leaks at other places.

Make sure that a spark occurs in each cylinder as follows: If magneto or battery with nonvibrating coil is used: Disconnect wire from sparkplug, hold end about one-eighth inch from cylinder or terminal of sparkplug. Have motor cranked briskly and see if spark occurs. Examine adjustment of interrupter points. See that wires are placed correctly and not short circuited. Take out sparkplug and lay it on the cylinder, being careful that base of plug only touches the cylinder and that ignition wire is connected. Have motor cranked briskly and see if spark occurs. Check timing of magneto and see that all brushes are making contact. See if there is gasoline in the carburetor. See that there is gasoline in the tank. Examine valve at tank. Prime carburetor and see that spray nozzle passage is clear. Be sure throttle is open. Prime cylinders by putting about a teaspoonful of gasoline in through pet cock or sparkplug opening. Adjust carburetor if necessary.

Tables of Engine Troubles.—The following tabulation was prepared and originated by the writer almost at the start of the automotive industry, over twenty years ago, to outline in a simple manner the various troubles and derangements that interfere with efficient internal-combustion engine action. This procedure has been widely adopted by engine builders and most instruction books contain similar instructions applying to the motor involved. The instructions given are general and apply to no specific make of engine. The parts and their functions are practically the same in all gas- or gasoline-engines of the four-cycle type, and the general instructions given apply just as well to all hydrocarbon engines, even if the parts differ in form materially. The essential components are clearly indicated in the many part sectional drawings in this book so they may be easily recognized. The various defects that may materialize are tabulated in a manner that makes for ready reference, and the various defective conditions are found opposite the part affected, and under a heading that denotes the main trouble to which the others are contributing causes. The various symptoms denoting the individual troubles outlined are given to facilitate their recognition in a positive manner.

Brief note is also made of the remedies for the restoration of the defective part or condition. It is apparent that a table of this character is intended merely as a guide, and it is a compilation of practically all the known troubles that may materialize in gas-engine operation. While most of the defects outlined are common enough to warrant suspicion, they will never exist in an engine all at the same time, and it will be necessary to make a systematic search for such of those as exist.

To use the list advantageously, it is necessary to know one main trouble easily recognized. For example, if the powerplant is noisy, look for the possible troubles under the head of Noisy Operation; if it lacks capacity,

the derangement will undoubtedly be found under the head of Lost Power. It is assumed in all cases that the trouble exists in the powerplant or its components, and not in the auxiliary members of the ignition, carburetion, lubrication, or cooling systems. The novice and student will readily recognize the parts of the average aviation engine by referring to the very complete and clearly lettered illustrations of mechanism given in many parts of this treatise.

LOST POWER AND OVERHEATING

PART AFFECTED	NATURE OF TROUBLE	SYMPTOMS AND EFFECTS	REMEDY
Water Pipe Joint.	Loose.	Loss of water, heating.	Tighten bolts, replace gaskets.
Sparkplug.	Leakage in threads, insulation, or packing.	Loss of power. Hissing caused by escaping gas.	Replace insulation if defective, screw down tighter.
Compression Release Cock.	Leak in threads. Leak in fitting.	Loss of power. Whistling or hissing.	Tighten if loose. Grind fitting to new seating in body.
Combustion-Chamber.	Crack or blowhole Roughness. Carbon deposits. Sharp edges.	Loss of compression. Preignition.	Fill by welding. Smooth out roughness. Scrape out or dissolve carbon.
Valve Chamber Cap.	Leak in threads. Defective gasket.	Loss of compression. Hissing.	Remove. Apply pipe compound to threads and replace. Use new gasket or packing.
Valve Head.	Warped. Scored or pitted. Carbonized Covered with scale. Loose on stem (two piece valves only.)	Loss of compression.	True up in lathe. Grind to seat. Scrape off. Smooth with emery cloth. Tighten by riveting.
Valve Seat.	Warped or pitted. Covered with carbon. Foreign matter between valve and seat.	Loss of compression.	Use reseating reamer. Clean off and grind valve to seat.
Valve Stem.	Covered with scale. Bent. Binding in guide. Stuck in guide.	Valve does not close. Loss of compression.	Clean with emery cloth; straighten. True up and smooth off, free with kerosene.
Valve Stem Guide.	Burnt or rough. Loose in valve chamber.	Valve may stick. Action irregular.	Clean out hole. Screw in tighter.
Valve Spring.	Weak or broken.	Valve does not close.	
Valve Operating Plunger.	Loose in guide. Too much clearance between valve stem	Valve action poor Lift insufficient.	Replace with new. Adjust screw closer.
Valve Lift Adjusting Screw.	Threads stripped. Too near valve. Too far from valve.	Poor valve action.	Replace with new. Adjust with proper reference to valve stem.
Valve Lift Cam.	Worn cam contour. Loose on shaft. Out of time.	Not enough valve lift Will not lift valve. Valve opens at wrong time.	Replace with new. Replace pins or keys. Set to open properly.

LOST POWER AND OVERHEATING—*Continued*

PART AFFECTED	NATURE OF TROUBLE	SYMPTOMS AND EFFECTS	REMEDY
Camshaft.	Sprung or twisted.	Valves out of time.	Straighten.
Camshaft Bushing.	Worn.	Not enough valve lift.	Replace.
Camshaft Drive Gear.	Loose on shaft. Out of time. Worn or broken teeth.	Irregular valve action.	Fasten securely. Time properly. Replace with new.
Cam Fastenings.	Worn or broken.	Valves out of time.	Replace with new.
Cylinder Wall.	Scored, gas leaks. Poor lubrication causes friction.	Poor compression. Overheating.	Grind out bore. Repair oiling system.
Piston.	Binds in cylinder. Walls scored. Worn out of round.	Overheating. Poor compression.	Lap off excess metal. Replace with new.
Piston Rings.	Loss of spring. Loose in grooves. Scored. Worn or broken. Slots in line.	Loss of compression. Gas blows by.	Peen ring or replace. Fit new rings. Grind smooth. Replace. Turn slots apart.
	Carbon in grooves. Insufficient opening. Binding on cylinder.	Overheating because of friction.	Remove deposits. File slot. Grind or lap to fit cylinder bore.
Wristpin.	Loose, scores cylinder.	Loss of compression.	Fasten securely. Replace cylinder if groove is deep.
Crankshaft.	Scored or rough on journals. Sprung.	Overheating because of friction.	Smooth up. Straighten.
Crank Bearings. Main Bearings.	Adjusted too tight. Defective oiling. Brasses burned.	Overheating because of friction.	Adjust freely, clean out oil holes and enlarge oil grooves.
Oil Sump.	Insufficient oil. Poor lubricant. Dirty oil.	Overheating.	Replenish supply. Use best oil. Wash out with kerosene; put in clean oil.
Water Space. Water Pipes.	Clogged with sediment or scale.	Overheating.	Dissolve foreign matter, remove.
Piston Head.	Cracked (rare). Carbon deposits.	Loss of compression. Preignition.	Weld by autogenous process. Scrape off carbon accumulations.

NOISY OPERATION OF POWERPLANT

PART AFFECTED	NATURE OF TROUBLE	CHARACTER OF NOISE	REMEDY
Compression Release Cock.	Leakage.	Hissing.	Previously given.
Sparkplug.	Leakage.	Hissing.	Previously given.
Valve Chamber Cap.	Leakage.	Hiss or whistle.	Previously given.
Combustion-Chamber.	Carbon deposits.	Knocking.	Previously given.
Inlet Valve Seat.	Defects previously given.	Popping in carburetor.	Previously given.
Valve Head.	Loose on stem.	Clicking.	Previously given.
Valve Stem. Valve Stem Guide.	Wear or looseness.	Rattle or clicking.	Previously given.
Inlet Valve.	Closes too late. Opens too early.	Blowback in carburetor.	Previously given.
Valve Spring.	Weak or broken.	Blowback in carburetor.	Previously given.

NOISY OPERATION OF POWERPLANT—*Continued*

PART AFFECTED	NATURE OF TROUBLE	CHARACTER OF NOISE	REMEDY
Cylinder Casting.	Retaining bolts loose. Piston strikes at upper end.	Sharp metallic knock.	Tighten bolts. Round edges of piston top.
Cylinder Wall.	Scored.	Hissing.	Previously given.
Valve Stem Clearance.	Too much. Too little (inlet valve).	Clicking. Blowback in carburetor.	Previously given.
Valve Operating Plunger. Plunger Guide.	Looseness.	Rattle or clicking.	Previously given.
Timing Gears.	Loose on fastenings. Worn teeth. Meshed too deeply.	Metallic knock. Rattle. Grinding.	Previously given.
Cylinder or Piston.	No oil, or poor lubricant.	Grinding.	Repair oil system.
Cam.	Loose on shaft. Worn contour.	Metallic knock.	Previously given.
Camshaft Bearing.	Looseness or wear.	Slight knock.	Previously given.
Cam Fastening.	Looseness.	Clicking.	Previously given.
Piston.	Binding in cylinder. Worn oval, causes side slap in cylinder.	Grinding or dull squeak. Dull hammering.	Previously given.
Piston Head.	Carbon deposits.	Knocking.	Previously given.
Piston Rings.	Defective oiling. Leakage. Binding in cylinder.	Squeaking. Hissing. Grinding.	Previously given.
Wristpin.	Loose in piston. Worn.	Dull metallic knock.	Replace with new member.
Connecting Rod.	Wear in upper bushing. Wear at crankpin. Side play in piston.	Distinct knock.	Adjust or replace. Scrape and fit. Use longer wristpin bushing.
Crank Bearings.	Looseness. Excessive end play. Binding, fitted too tight.	Metallic knock. Intermittent knock. Squeaking.	Refit bearings. Longer bushings needed. Insert shims to allow more play.
Main Bearings.	Looseness. Defective lubrication.	Metallic knock. Squeaking.	Fit brasses closer to shaft. Clean out oil holes and grooves.
Connecting Rod Bolts. Main Bearing Bolts.	Loose.	Sharp knock.	Tighten.
Crankshaft.	Defective oiling.	Squeaking.	Previously given.
Engine Base.	Loose on frame.	Sharp pounding.	Tighten bolts.
Lower Half Crankcase.	Bolts loose.	Knocking.	Tighten bolts.
Flywheel.*	Loose on crankshaft.	Very sharp knock.	Tighten retention bolts or fit new keys.
Oil Sump.	Oil level too low. Poor lubricant.	Grinding and squeak in all bearings.	Replenish with best cylinder oil.
Valve Plunger Retention Stirrups.	Looseness.	Clicking.	Tighten nuts.
Fan.	Blade loose. Blade strikes cooler.	Clicking or rattle.	Tighten. Bend back.
Exhaust Pipe Joints.	Leakage.	Sharp hissing.	Tighten or use new gasket.

NOISY OPERATION OF POWERPLANT—*Concluded*

PART AFFECTED	NATURE OF TROUBLE	CHARACTER OF NOISE	REMEDY
Crankcase Packing.	Leakage.	Blowing sound.	Use new packing. Tighten bolts.
Water Pipe.	Leaks. Loss of water. Clogged with sediment.	Pounding because engine heats.	Previously given.
Water Jacket.	Clogged with sediment. Walls covered with scale.	Knocking because engine heats.	Dissolve scale and flush out water space with water under pressure.

* Dirigible engines only. Loose propeller hub will cause the same noise.

"SKIPPING" OR IRREGULAR OPERATION

PART AFFECTED	NATURE OF TROUBLE	SYMPTOMS AND EFFECTS	REMEDY
Compression Relief Cock.	Leak in threads or spigot.	Dilutes mixture with air, causes blowback.	Screw down tighter. Grind spigot to seat with emery.
Sparkplug.	Leak in threads. Defective gasket. Cracked insulator. Points too near. Points covered with carbon. Too much air gap.	Dilutes mixture. Allows short circuit. No spark.	Screw down tighter. Replace with new. Set points .020" apart for magneto, .030" for battery spark.
Valve Chamber Cap.	Leak in threads. Defective gasket.	Dilutes mixture by allowing air to enter cylinder on suction stroke.	Previously given.
Combustion-Chamber.	Carbon deposits.	Preignition.	Scrape out.
Valve Head.	Warped or pitted. Loose on stem.	Dilutes charge with poor air or gas.	Previously given.
Valve Stem.	Binding in guide. Sticking.	Irregular valve action.	Previously given.
Valve Seat.	Scored or warped. Cracked. Covered with scale. Dirt under valve.	Gas leak, poor mixture. Poor compression. Valve will not close.	Previously given.
Induction Pipe.	Leak at joints. Crack or blowhole.	Mixture diluted with excess air.	Stop all leaks.
Inlet Valve.	Closes too late. Opens too early.	Blowback in carburetor.	Time properly.
Exhaust Valve.	Opens too late. Closes too early.	Retention of burnt gas dilutes charge.	Time properly.
Valve Stem Guide.	Bent or carbonized.	Causes valve to stick.	Previously given.
Inlet Valve Stem Guide.	Worn, stem loose.	Air drawn in on suction thins gas.	Bush guide or use new member.
Valve Spring.	Weakened or broken.	Irregular action.	Use new spring.
Valve Stem Clearance.	Too little. Too much.	Valve will not shut. Valve opens late, closes early.	Adjust gap .009" inlet, .010" exhaust.*
Valve Spring Collar Key.	Broken.	Releases spring.	Replace.

"SKIPPING" OR IRREGULAR OPERATION—*Continued*

PART AFFECTED	NATURE OF TROUBLE	SYMPTOMS AND EFFECTS	REMEDY
Cam.	Worn cam contour. Loose on shaft. Out of time.	Valve lift reduced. Does not lift valve. Valves operate at wrong time.	Previously given.
Camshaft Bearing.	Looseness or wear.	Valve timing altered. Valve lift decreased.	Replace.
Camshaft.	Twisted.	Valves out of time.	Previously given.
Cam Fastening.	Worn or broken.	Valve action irregular.	Replace with new.
Valve Operating Plunger.	Loose in guide.	Alters valve timing.	Replace with new.
Valve Plunger Guide.	Wear in bore. Loose on engine base.	Alters valve timing.	Replace or bush. Fasten securely.
Timing Gears.	Not properly meshed. Loose on shaft.	Valves out of time. Valves do not operate.	Retime properly. Fasten to shaft.
Piston.	Walls scored.	Leakage of gas.	Smooth up if possible.
Piston Head.	Carbon deposits. Crack or blowhole (rare).	Cause premature ignition.	Previously given.
Piston Rings.	No spring. Loose in grooves, worn or broken.	Leakage weakens suction.	Previously given.
Cylinder Wall.	Scored by wristpin. Scored by lack of oil.	Gas leaks by. Poor suction.	Previously given.

* This varies on different motors. Special instructions of motor builder to be followed in each case.

Ignition System Troubles Summarized

Motor Will Not Start or Starts Hard

Loose Battery Terminal.
Magneto Ground Wire Shorted.
Magneto Defective (No Spark at Plugs).
Broken Sparkplug Insulation.
Carbon Deposits or Oil Between Plug Points.
Sparkplug Points Too Near Together or Too Far Apart.
Wrong Cables to Plugs.
Short Circuited Secondary Cable.
Broken Secondary Cable.

Storage Battery Weak.
Storage Battery Discharged.
Poor Contact at Timer.
Timer Points Dirty.
} Battery Systems Only.

Poor Contact at Switch.
Primary Wires Broken, or Short Circuited.
Battery Grounded in Metal Container.
Battery Connectors Broken or Loose.
Timer Points Out of Adjustment.
Defects in Induction Coil.
} Battery and Coil Ignition System Only.

Ignition Timing Wrong, Spark Too Late or Too Early.
Defective Platinum Points in Breaker Box (Magneto).
Points Not Separating. (Battery Timer.)
Broken Contact Maker Spring.
No Contact at Secondary Collector Brush.
Platinum Contact Points Burnt or Pitted.
Contact Breaker Bell Crank Stuck.
Fiber Bushing in Bell Crank Swollen.
Short Circuiting Spring Always in Contact.
Dirt or Water in Magneto Casing.
Oil in Contact Breaker.
Oil Soaked Brush and Collector Ring.
Distril [illegible] h Carbon Particles.

Motor Stops Without Warning

Broken Magneto Carbon Brush.
Broken Lead Wire.
Battery Ignition Systems. } Broken Ground Wire.
Water on High Tension Magneto Terminal.
Main Secondary Cable Burnt Through by Hot Exhaust Pipe (Transformer Coil or Magneto Systems With Separate Distributors).
Particle of Carbon Between Sparkplug Points.
Magneto Short Circuited by Ground Wire.
Magneto Out of Time, Due to Slipping Drive.
Water or Oil in Safety Spark Gap (Multi-Cylinder Magneto).
Magneto Contact Breaker or Timer Stuck in Retard Position.
Worn Fiber Block in Magneto Contact Breaker.
Binding Fiber Bushing in Contact Breaker Bell Crank.
Spark Advance Rod or Wire Broken.
Contact Breaker Parts Stuck.

Motor Runs Irregularly or Misfires

Loose Wiring or Terminals.
Broken Sparkplug Insulator.
Sparkplug Points Sooted or Oily.
Wrong Spark Gap at Plug Points.
Leaking Secondary Cable.
Prematurely Grounded Primary Wire.
Batteries Running Down (Battery Ignition Only).
Poor Adjustment of Contact Points at Timer.
Wire Broken Inside of Insulation.
Loose Platinum Points in Magneto.
Weak Contact Spring.
Broken Collector Brush.
Dirt in Magneto Distributor Casing or Contact Breaker.
Worn Fiber Block or Cam Plate in Magneto.
Worn Cam or Contact Roll in Timer (Battery System Only).

Dirty Oil in Timer.
Sticking Coil Vibrators.
Coil Vibrator Points Pitted.
Oil Soaked Magneto Winding.
Punctured Magneto or Coil Winding.
Distributor Contact Segments Rough.
Sulphated Storage Battery Terminals.
Weak Magnets in Magneto.
Poor Contact at Magneto Contact Breaker Points.

Electrical System Components.—To further simplify the location of electrical system faults it is thought desirable to outline the defects that can be present in the various parts of the individual devices comprising the ignition system. If an airplane engine is provided with magneto ignition solely, as most engines are at the present time, no attention need be paid to such items as storage or dry batteries, timer or induction coil. There seems to be some development in the direction of battery ignition so it has been considered desirable to include components of these systems as well as the almost universally used magneto group. Sparkplugs, wiring and switches are needed with either system.

SPARKPLUGS

DEFECT	TROUBLE CAUSED	REMEDY
Insulation cracked.	Plug inoperative.	New insulation.
Insulation oil soaked.	Cylinder misfires.	Clean.
Carbon deposits.	Short circuited spark.	Remove.
Insulator loose.	Cylinder misfires.	Tighten.
Gasket broken.	Gas leaks by.	New gasket.
Electrode loose on shell.	Cylinder misfires.	Tighten.
Wire loose in insulator.	Cylinder misfires.	Tighten.
Air gap too close.	Short circuits spark.	Set correctly.
Air gap too wide.	Spark will not jump.	Set points .015" to .030" apart.
Loose terminal.	Cylinder may misfire.	Tighten.
Plug loose in cylinder.	Gas leaks.	Tighten.
Mica insulation oil soaked.	Short circuits spark.	Replace.

MAGNETO

DEFECT	TROUBLE CAUSED	REMEDY
Dirty oil in distributor.	Engine misfires.	Clean.
Metal dust in distributor.	Engine misfires.	Clean.
Brushes not making contact.	Current cannot pass.	Strengthen spring.
Distributor segments worn.	Engine misfires.	Secure even bearing.
Collecting brush broken.	Engine misfires.	New brush.
Distributing brush broken.	Engine misfires.	New brush.
Oil soaked winding.	Engine misfires.	Clean.
Magnets loose on pole pieces.	Engine misfires.	Tighten screws.
Armature rubs.	Engine misfires.	Repair bearings.
Bearings worn.	Noisy.	Replace.
Magnets weak.	Weak spark.	Recharge.
Contact breaker points pitted.	Engine misfires.	Clean.
Breaker points out of adjustment.	Engine misfires.	Reset.
Defective winding (rare).	No spark.	Replace.
Punctured condenser (rare).	Weak or no spark.	Replace.

Driving gear loose.	Noise.	Tighten.
Magneto armature out of time.	Spark will not fire charge.	Retime.
Magneto loose on base.	Misfiring and noisy.	Tighten.
Contact breaker cam worn.	Misfiring.	Replace.
Fiber shoe or rolls worn (Bosch).	Misfiring.	Replace.
Fiber bushing binding in contact lever (Bosch).	Misfiring.	Ream slightly.
Contact lever return spring broken.	No spark.	Replace.
Contact lever return spring weak.	Misfiring.	Replace.
Ground wire grounded.	No spark.	Insulate.
Ground wire broken.	Engine will not stop.	Connect up.
Safety spark gap dirty.	No spark.	Clean.
Fused metal in spark gap.	No spark.	Remove.
Safety spark gap points too close.	Misfiring.	Set properly.
Loose distributor terminals.	Misfiring.	Tighten.
Contact breaker sticks.	No spark control.	Remove and clean bearings.
Magneto switch short-circuited.	No spark.	Insulate.
Magneto switch open circuit.	Engine will not stop.	Restore contact.

STORAGE BATTERY

DEFECT	TROUBLE CAUSED	REMEDY
Electrolyte low.	Weak current.	Replenish with distilled water.
Loose terminals.	Misfiring.	Tighten.
Sulphated terminals.	Misfiring.	Clean thoroughly and coat with vaseline.
Battery discharged.	Misfiring or no spark.	New charge.
Electrolyte weak.	Weak current.	Bring to proper specific gravity.
Plates sulphated.	Poor capacity.	Special slow charge.
Sediment or mud in bottom.	Weak current.	Clean out.
Active material loose in grids.	Poor capacity.	New plates.
Moisture or acid on top of cells.	Shorts terminals.	Remove.
Plugged vent cap.	Buckles cell jars.	Make vent hole.
Cracked vent cap.	Acid spills out.	New cap.
Cracked cell jar.	Electrolyte runs out.	New jar.

TIMER

DEFECT	TROUBLE CAUSED	REMEDY
Contact segments worn or pitted.	Misfiring.	Grind down smooth.
Platinum points pitted.	Misfiring.	Smooth with oil stone.
Dirty oil or metal dust in interior.	Misfiring.	Clean out.
Worn bearing.	Misfiring.	Replace.
Loose terminals.	Misfiring.	Tighten.
Worn revolving contact brush.	Misfiring.	Replace.
Out of time.	Irregular spark.	Reset

INDUCTION COIL

DEFECT	TROUBLE CAUSED	REMEDY
Loose terminals.	Misfiring.	Tighten.
Broken connections.	No spark.	Make new joints.
Vibrators out of adjustment.*	Misfiring.	Readjust.
Vibrator points pitted.*	Misfiring.	Clean.
Defective condenser } rare. Defective winding }	No spark.	Send to maker for repairs.

Poor contact at switch.	Misfiring.	Tighten.
Broken internal wiring.	No spark.	Replace.

* Booster or starting coils only. Ignition coils have no vibrator.

WIRING

DEFECT	TROUBLE CAUSED	REMEDY
Loose terminals anywhere.	Misfiring.	Tighten.
Broken plug wire.	One cylinder will not fire.	Replace.
Broken timer wire.	No spark.	Replace.
Broken main battery wire. Broken battery ground wire.	No spark.	Replace.
Broken magneto ground wire.	Engine will not stop.	Replace.
Chafed insulation anywhere. Short circuit anywhere.	Misfiring.	Insulate.

Carburetion System Faults Summarized

Motor Starts Hard or Will Not Start

No Gasoline in Tank.
No Gasoline in Carburetor Float Chamber.
Tank Shut-Off Closed.
Clogged Filter Screen.
Fuel Supply Pipe Clogged.
Gasoline Level Too Low.
Gasoline Level Too High (Flooding).
Bent or Stuck Float Lever.
Loose or Defective Inlet Manifold.
Defective Inlet Manifold Packing.
Not Enough Gasoline at Jet.
Cylinders Flooded With Gas.
Fuel Soaked Cork Float (Causes Flooding).
Water in Carburetor Spray Nozzle.
Dirt in Float Chamber.
Gas Mixture Too Lean.
Carburetor Frozen (Winter Only).

Motor Stops in Flight

Gasoline Shut-Off Valve Jarred Closed.
Gasoline Supply Pipe Clogged.
No Gasoline in Tank.
Spray Nozzle Stopped Up With Dirt.
Water in Spray Nozzle.
Air Lock in Gasoline Pipe.
Broken Air Line or Leaky Tank (Pressure Feed System Only).
Fuel Supply Pipe Partially or Wholly Clogged.
Air Vent in Tank Filler Cap Stopped Up (Gravity Feed System).
Float Needle Valve Stuck.
Water or Dirt in Spray Nozzle.
Mixture Adjusting Needle Jarred Loose (Rotary Motors Only).

Fuel Filter Screen Clogged with Dirt or Trap Filled with Water.
Ice in Fuel Lines (Winter Only).
Ice In Air Intake. (Winter Only.)

Motor Races, Will Not Throttle Down

Air Leak in Inlet Piping.
Air Leak Through Inlet Valve Guides.
Control Rods Broken, Throttle Stuck Open.
Defective Induction Pipe Joints.
Leaky Carburetor Flange Packing.
Throttle Not Closing, Stuck Control.
Poor Slow Speed Adjustment.

Motor Misfires

Carburetor Float Chamber Getting Dry.
Water or Dirt in Gasoline.
Poor Gasoline Adjustment (Rotary Motors).
Not Enough Gasoline in Float Chamber.
Too Much Gasoline, Carburetor Flooding.
Incorrect Jet or Choke.

Noisy Operation

Incorrectly Timed Inlet Valves. Inlet Valves Not Seating. Defective Inlet Valve Springs. Dirt Under Inlet Valve Seat.	Popping or Blowing Back in Carburetor.
Mixture Not Exploding Regularly. Exhaust Valves Sticking. Dirt Under Exhaust Valve Seat.	Muffler or Manifold Explosions.

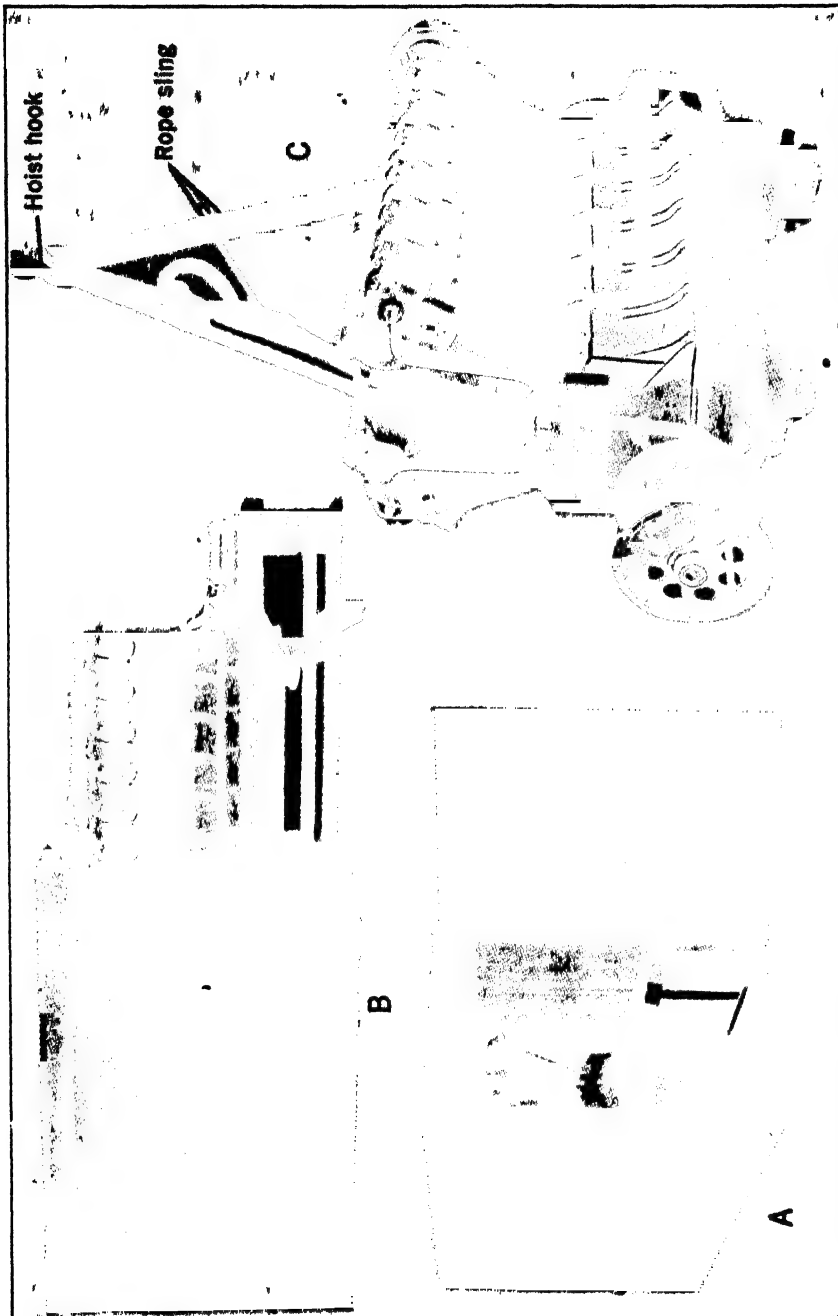


Fig. 437.—Illustration Showing Method of Removing Liberty-12 Airplane Engine from the Packing Case and How to Sling it Without Damaging the Engine.

CHAPTER XXVII

INSTALLING, OPERATING AND REPAIR OF LIBERTY MOTORS

Unpacking—Engine Bed—Water Piping—Oil Piping—Gasoline Piping—Controls—Propeller Mounting—Pitch of Propeller—Preparing Engine for Service—Fill Cooling System—Fill Oiling System—Properties of Oils—Instructions for Starting Engine—Cold Weather Suggestions—Liberty Engine Troubles—Periodic Inspection—Overhaul and Repair—Electrical Equipment—Generator—Switch—Distributors—High Tension Wiring—Electrical System Inspection Procedure—Voltage Regulator—Ignition Switch—Preparing Battery for Service—Water Outlet Headers—Camshaft Housing Units—Lower Camshaft Drive Shafts—Generator Driving Shaft Assembly—Liberty 12 Oiling System—Oil Pump—Cooling System—Disassembling and Inspection—Water Pump Bevel Driver—Cylinder Assembly—Remove the Valves—Dismounting Pistons—Rings—Connecting Rods—Crankshaft—Removing Propeller Hub—Fitting Propeller Hub—Crankcase—To Assemble the Engine—Crankcase Lower Half—Pistons and Cylinders—Outlet Headers—Carburetors—Timing Engine—Tappet Gap and Firing Point—Synchronizing Breakers—Water Inlet Manifolds—Oil Pump Assembly—Crankcase Breathers—Testing—Summary of Clearances.

Each original case contained one engine, one oil gauge, one air gauge, one battery, one tool roll with tools, one tachometer with shaft and connection, one box of spare parts for electrical equipment, one instruction book and various spare engine parts. Fastened on the inside of the removable end of the case will be found a list of all these parts.

Unpacking.—Remove all lag screws through sides and ends of case as shown at Fig. 437 A. 2. Remove wood screws from marked end of case. 3. Take out this end of the case and remove the accessories. 4. The engine, with the benches or skids to which it is bolted, may now be drawn out of the open end of the case on to the floor, as at Fig. 437 B. 5. Take out the bolts by means of which the engine is fastened to the skids and hoist the engine with a rope sling arranged as shown in Fig. 437 C. See that the rope is pushed snugly back against the crankcase at the pump end and is passed on the outside of generator and throttle shaft and inside of altitude control rod. Block out the rope at the propeller end of the engine with a piece of wood two inches by four inches so that it will clear the oil pipes. Use a rope long enough to make a sling of about the length shown.

Engine Bed.—The engine supporting members should be not less than 14⅞ inches inside to inside. The engine is held down by means of fourteen ⅜ inch bolts spaced as shown in the Installation Diagram, Fig. 438. These bolts should be provided with liberal washers under the heads and the nuts should be castellated for cotterpins or lock wires.

Sufficient radiator area should be provided to hold the water temperature at not to exceed 200 degrees Fahrenheit with the radiator set in the propeller draft. The radiator should be fitted with adjustable shutters or an equivalent method of maintaining the proper water temperature in cold weather or at high altitudes. The water temperature should not be allowed to become lower than 160 degrees Fahrenheit or carburetion will be affected and burned exhaust valves result.

Water Piping.—All water piping from radiator to engine and engine to radiator should have an inside diameter of not less than 1⅞ inches and an outside diameter of two inches The piping should be metal bent to a shape which will permit easy bends of the greatest possible radius. All unnecessary bends should be avoided. Air pockets should be avoided, but should the installation be such that an air pocket is unavoidable, a vent cock should be fitted at the highest point. The metal tubing should be of such a length that, where it is attached to the engine, and radiator, by means of rubber hose connections, not more than one-half of an inch of rubber hose will be exposed to the water. The ends of the tubing over which the hose connections are slipped, should be corrugated Hose connections should be taped and shellacked over the tape. Hose clamps should be bent to fit snugly over the shellacked tape and should be drawn up only tightly enough to prevent leaks.

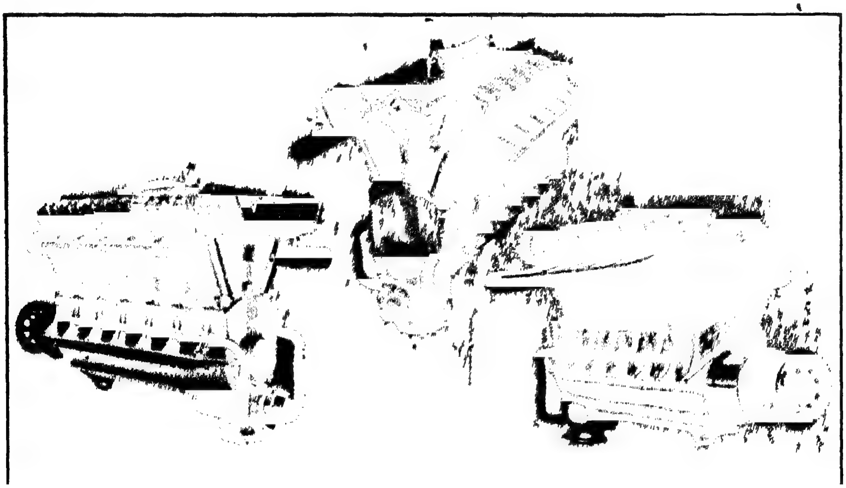


Fig. 437D.—Views of Liberty-12 Engine After Removal from Packing Case.

Oil Piping.—The instructions for the installation of water piping apply also to the oil piping except that special hose connections should be used so made that the inside layer is fabric instead of rubber. The surface exposed to the oil between the ends of the tubing should not exceed one-quarter of an inch in length.

Gasoline Piping.—Should be annealed copper tubing. This tubing to be of not less than ⅜ inch inside diameter from the tank to the T between the carburetors. A gasoline strainer or trap with a flow capacity of not less than 50 gallons per hour should be installed between the gasoline tank and the T. From this T to each carburetor the inside diameter of the tubing should be not less than $\frac{9}{32}$ inch. Piping should be clipped to the fuselage in such a manner as to reduce vibration to a minimum. Where it is fastened to two adjacent members, one rigid and one free to vibrate, or to two members which have a different period of vibration, a rubber hose connection should be interposed. This hose connection should be special and so made as to present a fabric surface to contact with the gasoline.

The ends of the copper tubing to be connected by means of the rubber hose should not be farther apart than one-quarter of an inch. Air pressure piping should be annealed copper tubing not less than $\frac{3}{16}$ inch inside diameter and installed in the same manner as the gasoline piping.

Spark, Throttle and Altitude Adjustment Controls.—Distributors and carburetors are so mounted that all control connections are made at the distributor end of the engine and the installation is such that motion at right angles to the center line of the engine is required to operate them.

The spark control requires $1\frac{5}{8}$ inches motion. The throttle control requires $1\frac{7}{8}$ inches motion. The altitude control requires $1\frac{1}{8}$ inches motion. Each control should be provided with a ratchet working over a toothed sector to hold it in any desired position. All controls should work freely and, at the same time, permit a minimum of lost motion.

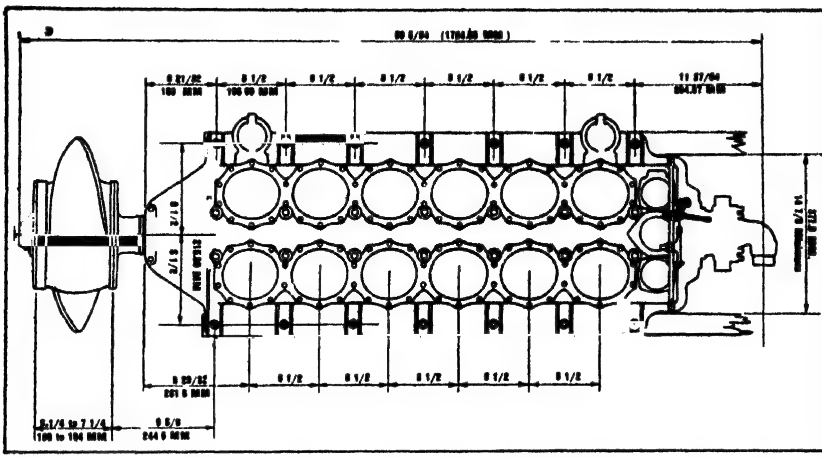


Fig. 438.—Installation Drawing Showing Spacing of Liberty Engine Hold-Down Bolts.

Electric Wiring.—All low tension wire should be No. 14 stranded cable for distances of ten feet or less, or No. 10 for distances up to 25 feet, well insulated with rubber and braid. Wire should be clipped to the fuselage at close intervals and should be taped and shellacked under the clips and wherever exposed to oil. All terminal nuts should be castellated and cotter pinned or screwed down on lock washers. The battery should be clamped rigidly in place in the fuselage. A battery of greater capacity than the standard one specified—part No. 8417—should not be connected to the generator or damage to the generator will result. The primary wiring is clearly shown in diagram at Fig. 439.

Tachometer Drive.—Tachometer is to be driven from a worm shaft incorporated in the base of the generator and parallel with the crankshaft. An angle adapter is provided which will permit running the flexible tachometer drive shaft at right angles to the crankshaft, if it is deemed advisable. The speed of this drive shaft is one-half crankshaft speed.

Air Pump and Gun Synchronizer.—Provision is made for mounting a mechanically driven air pump on the distributor end of the engine crankcase and driving it by means of a splined shaft fitting into the crankshaft

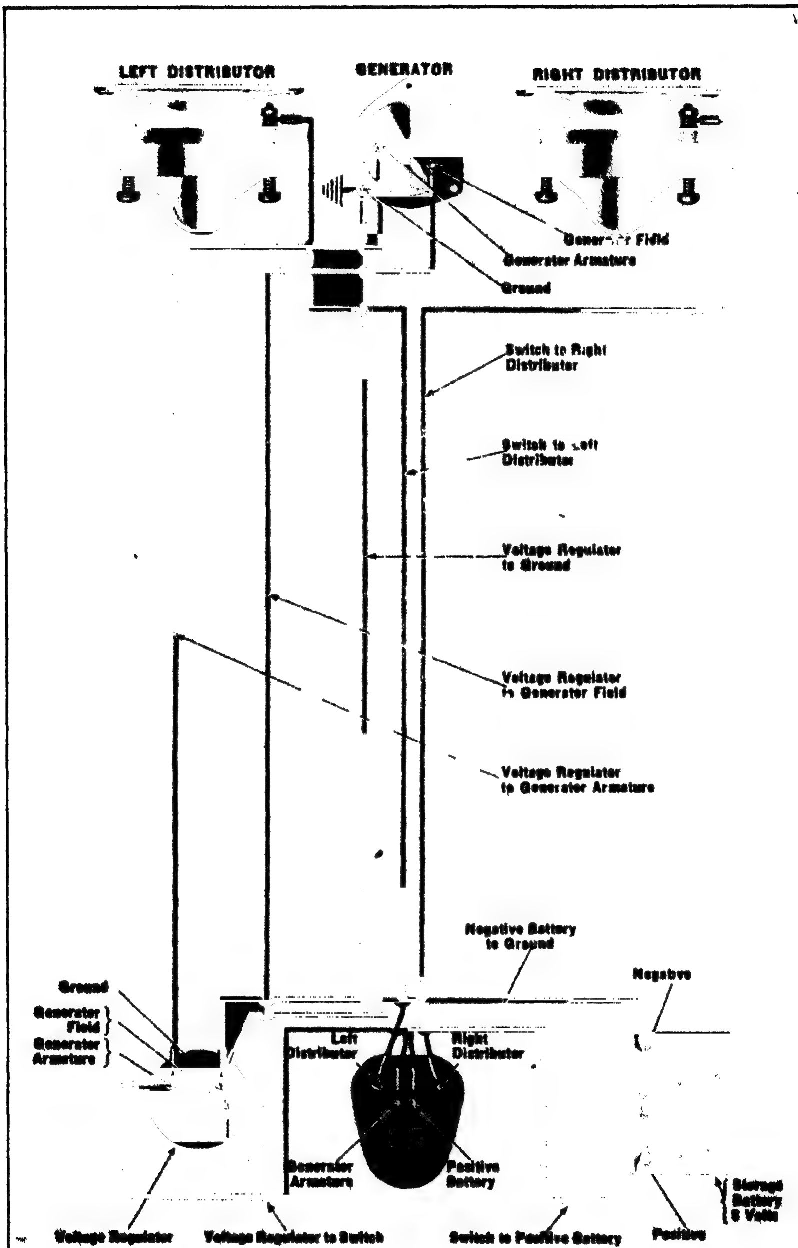


Fig. 439.—Wiring Diagram Showing Connection of Low-Tension Wiring of the Liberty Engine Ignition System.

gear. An extension of this shaft carries a double adjustable cam designed to operate a machine gun. The over-all length of the unit is six inches. It will be installed by the plane manufacturer.

Propeller Mounting.—Liberty "12" engines require a propeller which will permit a ground speed of not less than 1,500 r.p.m. and a flying speed of 1,650 to 1,700 r.p.m.[*] The propeller should not be more than nine feet in diameter. The blades should be nine inches or less in width and should narrow down at the tips. If a nose radiator is employed, that part of the blade which swings in front of the radiator should be designed with no effective pitch. The drawing at Fig. 440 shows a typical design working drawing of a propeller suitable to absorb the power of the Liberty engine.

Track of Propeller.—In mounting propellers, great care should be taken that both blades rotate in the same plane. This is termed the "track" of the propeller and may be checked as shown in Fig. 441 A. Measure the distance from the edge of one blade to some fixed point on the plane. This measurement should be made in a line parallel with the center line of the crankshaft. Now turn the propeller through 180 degrees or one-half revolution and measure the other blade in the same manner and the same distance from the center. The variation should not exceed 1/8 inch. The propeller can be trued up as to track by shimming between the hub flange and propeller with paper, brass shim stock, or, in extreme cases, by dressing off the face of the propeller where it comes in contact with the hub flange.

Pitch of Propeller.—The pitch of both blades should be checked with a protractor and level as shown in Fig. 441 B. Both blades should be checked at the same distance from the center of the propeller, or about two-thirds of the distance from center to tip. The difference in pitch between the two blades must not exceed 1/16 inch in nine inches. Variation in pitch can be corrected in the same manner as error in track, i.e., by shimming or by dressing off the propeller. The fitting of the propeller hub, and method of removing it are dealt with later.

Preparing Engine for Service.—While every possible precaution was taken by the builders at the factory to insure Liberty engines being properly assembled, the human element, which is not infallible, must always be considered, therefore: Inspect all visible bolts and nuts. See that they are properly drawn up and securely locked. Inspect propeller mounting. See that hub flange bolts are properly drawn up and securely locked. See that the retaining nut and lock nut are drawn up tight and be sure that the tongue of the lock wire passes through both. Check pitch and track of propeller. Inspect throttle and spark controls. See that throttles of both carburetor assemblies are synchronized. See that throttle control at pilot's seat permits full throttle opening. See that spark control at pilot's seat permits the specific range of advance and retard at the distributor. (Ten degrees after dead center—retarded, and 30 degrees before dead center—advanced).

Check tappet gap. With the cylinder set on the firing point, the gap between the inlet valve tappets and the valve stems should be .014 inch to .016 inch and between the exhaust valve tappets and valve stems should be .019 inch to .021 inch. Inspect all electrical connections. See that all wire terminals are properly soldered, clean and firmly attached at the dis-

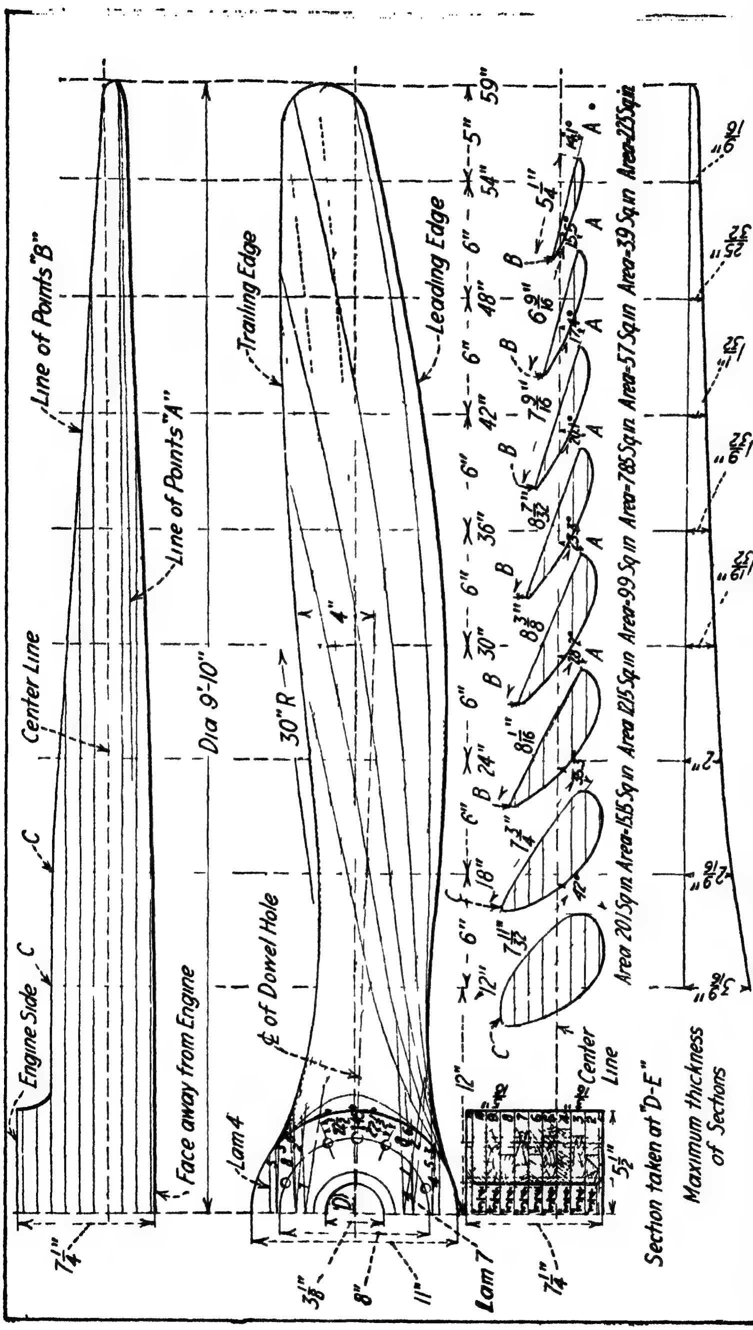


Fig. 440.—Working Drawing of a Wood Propeller Suitable for Liberty Engine, 400 Horsepower, Showing Dimensions and Sections of Blades at Various Stations.

tributors, generators, battery, switch and voltage regulator. All wire terminal nuts should be cotter pinned or screwed down on lock washers. See that all wires are properly insulated and supported at close intervals in a manner that the insulation will not be abraded.

Inspect Ignition System.—Note whether or not the mark on the distributor assembly base coincides with the corresponding mark on the camshaft housing flange. Remove distributor head assembly by unscrewing two composition nuts "B" and releasing four spring clips "A" as shown at Fig. 442. Check each breaker individually by turning engine over until breaker is wide open and testing "gap" (width of opening) by means of thickness gauge marked "distributor contacts" attached to distributor wrench. Gap should be .010 inch to .013 inch. The middle circuit breaker must open before the two main breakers for normal direction of rotation and should not close too soon after they open. Check this point by turning the engine over until one of the narrow lobes of the cam is directly under the block of the middle breaker arm when the main breakers should be just opening. Check timing of one main breaker on each distributor assembly with an eight volt lamp and battery or with an electric torch. Check synchronization of two distributor assemblies. Check inside of distributor covers by wiping with soft clean cloth moistened with alcohol or gasoline—dry carefully.

Replace covers so that terminal marked 1L is just to the left of the red mark on the assembly bases—spark retarded. With the engine set on the firing point of No. 1L cylinder; in other words, with the No. 1 crank set ten degrees past the compression dead center, the carbon brush in the end of the distributor rotor should bear on the brass contact marked 1L on the distributor head. Remove the sparkplugs and inspect carefully for defective or broken porcelain. The electrode should be tight in the insulator and the insulator should be properly gasketed and drawn up tightly to prevent gas leakage. Hot gas blowing through a plug will overheat and render it inoperative. A sparkplug which has been used and has given satisfactory service is always safer than a new plug, therefore, do not discard sparkplugs simply because they have been used, unless it is known positively that they are defective. Defects in sparkplugs will be most apparent when the plugs are hot. Check sparkplug "gap"—should be .015 inch to .018 inch. A gauge is provided marked "sparkplug" on the distributor wrench. Clean plugs—use a stiff brush and gasoline. Replace plugs—being careful that gasket is in place and that plug is drawn down tightly on it.

Trace out high tension wires from the distributors and be sure that each plug is connected to the correspondingly marked terminal on the distributor. Plugs on the side of the cylinders toward the propeller are connected to the left hand distributor. Plugs on the opposite side of the cylinders are connected to the right hand distributor as shown in diagram at Fig. 443. Order of firing—Standing at the distributor end of the engine and locking toward the propeller, the groups of cylinders are designated as "Left" and "Right" respectively and are numbered 1, 2, 3, 4, 5 and 6

beginning at the distributor end. The order of firing is as follows:

1	2	3	4	5	6	7	8	9	10	11	12
1L	6R	5L	2R	3L	4R	6L	1R	2L	5R	4L	3R

Fill the gasoline system. Gasoline of the following specifications is recommended: Specific gravity—58 to 65 Baumé. Initial boiling point—

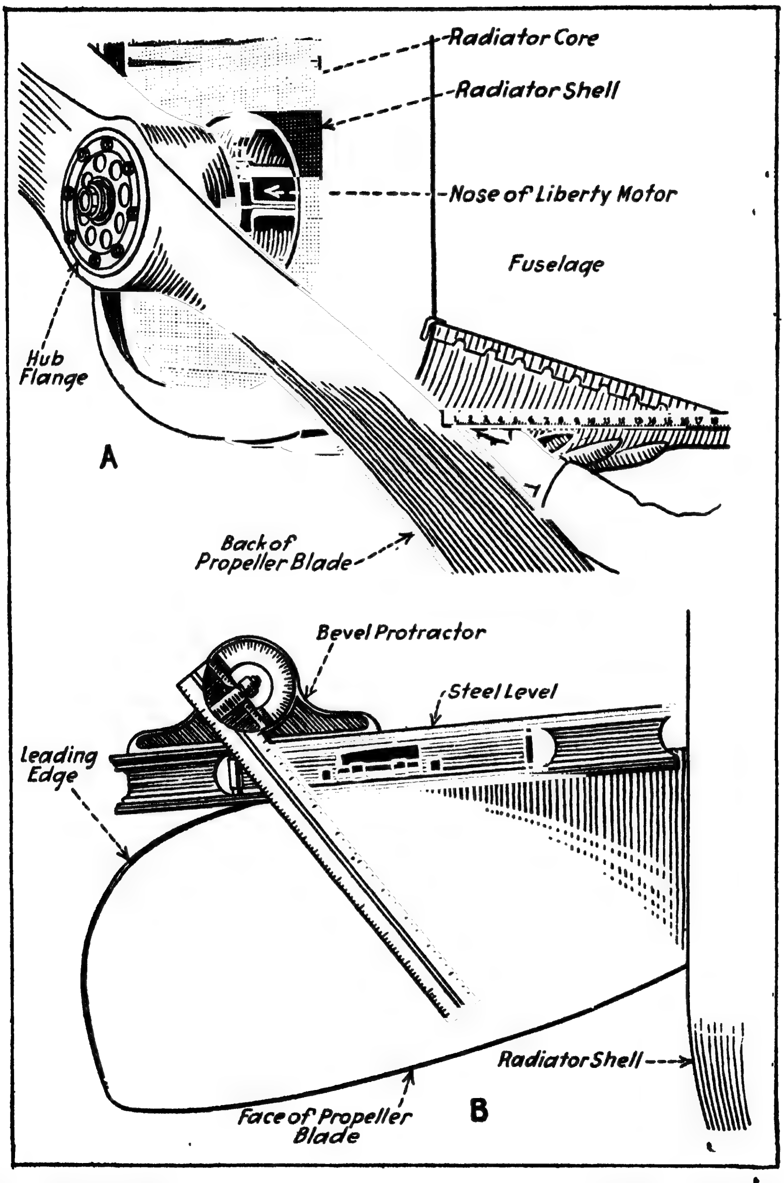


Fig. 441.—Drawing at A Shows Method of Testing Liberty Propeller for Track. The Use of Bevel Protractor for Testing the Pitch of the Blades is Shown at B.

102 degrees Fahrenheit—not higher than 120 degrees Fahrenheit. Final boiling point—350 degrees Fahrenheit. In filling the tank, pour the gasoline through a chamois skin to free it from water and impurities, but in doing so, make sure that the side of the funnel makes a firm contact with the side of the gasoline tank. Gasoline and chamois, when brought into contact, form static electricity which may cause a spark unless the funnel is grounded to the tank. Another method is to use a funnel with a very fine mesh brass wire screen. The chamois filters out foreign matter the wire mesh will allow to pass through it.

Pump up pressure on gasoline tank with hand pump until gauge shows three pounds. It is advisable to flush out the gasoline line on a new installation which is being filled for the first time. The gasoline pipes should have been left disconnected at the carburetor end until after filling the tank. The line may be washed out by turning on the stop cock at the tank and allowing a small quantity of gasoline to run through. Connect pipes, and, with gasoline shut-off cock open—Inspect all piping and connections carefully for leaks. Inspect gasoline strainer for leaks. Be sure that carburetor float chamber fills properly and that carburetor does not flood. To determine whether or not the carburetor float chamber is full, unscrew the cap over the needle valve. If the chamber is full the needle will be down on its seat and cannot be depressed further. If this needle can be pressed down, it would indicate either a stoppage in the pipe or insufficient pressure in the tank. Close gasoline shut-off cock.

Fill the Cooling System.—Use water which is as free from lime and other impurities as it is possible to obtain. It is assumed that the directions given previously for the installation of the cooling system have been carefully followed and that the piping is free from air pockets. Should there be an unavoidable air pocket in the line, open the air vent cock which should be provided at this point, and allow it to remain open during the process of filling until water flows freely from it. Examine the radiator, pump, water jackets, piping and all connections carefully to be sure that there are no leaks in the cooling system.

Fill Oiling System.—Lubricating oil of the following properties is recommended: Classification—High specific gravity oils:—This class includes all oils having a specific gravity above 0.9100 (or below 24 degrees Baumé conversion by the Tagliabue Manual, 9th edition, or below 23.85 degrees Baumé conversion by the Bureau of Standards' conversion table, Circular No. 57) and having a pour test below fifteen degrees Fahrenheit. Low specific gravity oils:—This class includes all oils having a specific gravity below 0.9100 (or above 24 degrees Baumé conversion by the Tagliabue Manual, 9th edition, or above 23.85 degrees Baumé conversion by the Bureau of Standards' conversion table, Circular No. 57) and having a pour test above fifteen degrees Fahrenheit. (Tested by the method of the American Society for Testing Materials.)

Physical Properties and Tests.—The oil must be made from pure, highly refined petroleum products, and must be suitable in every way for the entire lubrication of stationary cylinder aircraft engines operating under all conditions. The oil must be neutral in action and must not show the presence of moisture, sulphonates, soap, resin, or tarry constituents which would

indicate adulteration or lack of proper refining.

The viscosity of the oil, when tested in a Saybolt Universal Viscosimeter at 212 degrees Fahrenheit, shall be as follows:

High specific gravity oil ....70 seconds to 75 seconds.
Low specific gravity oil ....85 seconds to 90 seconds.

Pour Test:—The oil must pass the following pour test:

High specific gravity oil ....not over 15 degrees Fahrenheit.
Low specific gravity oil ....not over 40 degrees Fahrenheit.

Flash Point:—The oil must have a flash point over 350 degrees Fahrenheit in a Cleveland open cup. Carbon:—The oil must not show a carbon residue of over 1.5 per cent by the Conradson method. The carbon shown must be loose and flaky and must break up easily in the crucible. Emulsion Test:—One ounce of oil shall be placed in a standard four-ounce sample bottle with one ounce of distilled water. The mixture shall be heated to a temperature of 180 degrees Fahrenheit, and then shaken vigorously for five minutes. After standing for one hour, the oil must be clear and of the same color as before the test. All of the water must have settled and appear only slightly cloudy. All tests must be made in accordance with methods adopted by the American Society for Testing Materials. Detailed descriptions of the Conradson Carbon Test and the Pour Test have been reprinted in Signal Corps Specifications No. 3,525, which will be furnished on application to the War Department. Physical properties and tests to be determined as follows: Gravity, Baumé, at 60 degrees Fahrenheit. Flash, Cleveland open cup. Fire, Cleveland open cup. Viscosity, Saybolt Universal Viscosimeter, at 100, 130, and 212 degrees Fahrenheit. Pour Test, American Society for Testing Materials' method. Carbon, Conradson method. Color, Lovibund.

Remove plug No. 161 on side of oil pump housing during the process of filling the reservoir and allow it to remain out until oil flows from it. Do not fill oil reservoir more than three-quarters full. Fill oil piping with oil in the following manner: Incline engine so that propeller end is slightly higher than distributor end. Remove plug which closes one end of oil distributor pipe and fill this pipe with oil. Remove plugs in the camshaft housing cover plates nearest the propeller and pour about a pint of oil into each one. The above precaution will insure delivery of oil to the bearings immediately the engine is started.

See that Ignition Switches are in "OFF" position. Try compression of each cylinder separately by cranking the engine over slowly by means of the propeller and "rocking" it up to each compression point. Any weak cylinders can readily be detected, either by the decreased resistance to cranking or by the hissing sound due to a leaking valve or sparkplug. Locate the weak cylinder in the following manner; Crank the engine over again, meanwhile watching the No. 6L exhaust valve. When this valve is wide open the piston in No. 1L cylinder will be coming up on its compression stroke and just before the valve closes No. 1L will have reached its point of highest compression or top dead center. Now start from this point and crank the engine slowly past each compression period at the same

time calling off the numbers of the cylinders in their order of firing until the weak one is reached. For the causes of loss of compression see instructions to follow. It sometimes happens than an engine will show uneven compression when cold but will be all right at running temperature, so that it is not advisable to assume that anything is radically wrong with it until it has been warmed up and the compression tested again.

Instructions for Starting Engine.—Before starting a new engine or one which has stood idle for some time, it is advisable to inject a small quantity of lubricating oil (about one-half ounce) through a sparkplug hole in each cylinder. With the ignition switches "Off," turn the propeller forward through five or six revolutions to distribute the oil over the cylinder walls. Block wheels securely. Set throttle just slightly open, in other words, at

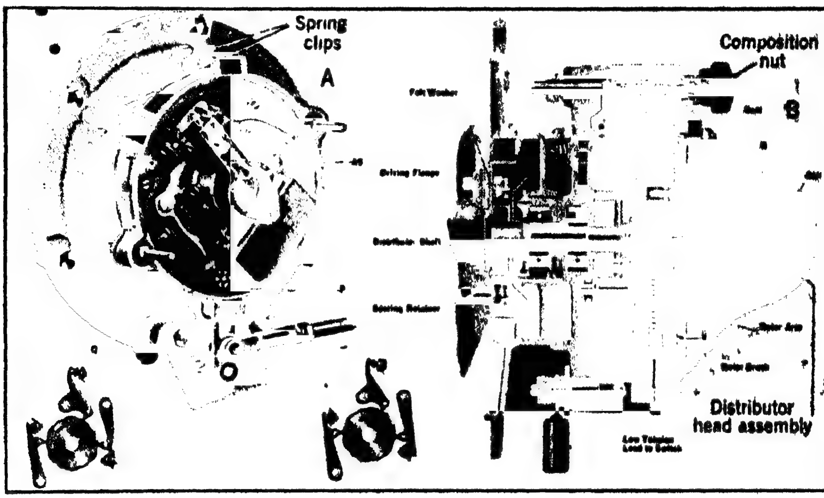


Fig. 442.—Illustrations Showing Construction of the Liberty Engine Distributor Head. View at the Left Shows the Distributor Head Assembly Removed to Show Breaker Cam Arrangement. View at Right is a Longitudinal Sectional Elevation.

a point which will run the engine at 600 to 800 r.p.m. Set spark at fully retarded position. The ignition system for Liberty engines is so designed as to absolutely prevent the production of a spark when turned backward, nor will the engine "kick back" if it should happen to "rock" after cranking. However, it is essential that the spark be retarded when cranking.

Prime engine by injecting a small quantity (fill priming cock twice) of gasoline through each priming cock. In cold weather it will be necessary to prime the engine a little more heavily than in warm weather. It is better, however, to insufficiently prime it than to prime it too heavily. With the ignition switch still "Off" turn engine forward two revolutions. Turn one (either one) ignition switch "on" and start engine by pulling steadily down on the propeller blade and at the same time away from it. The switch is so designed that, with both switches turned "on" the generator is connected in, which will result in a rather high rate of discharge from the battery and possible difficulty in starting. Both switches should be turned "on" however, as soon as the engine is running.

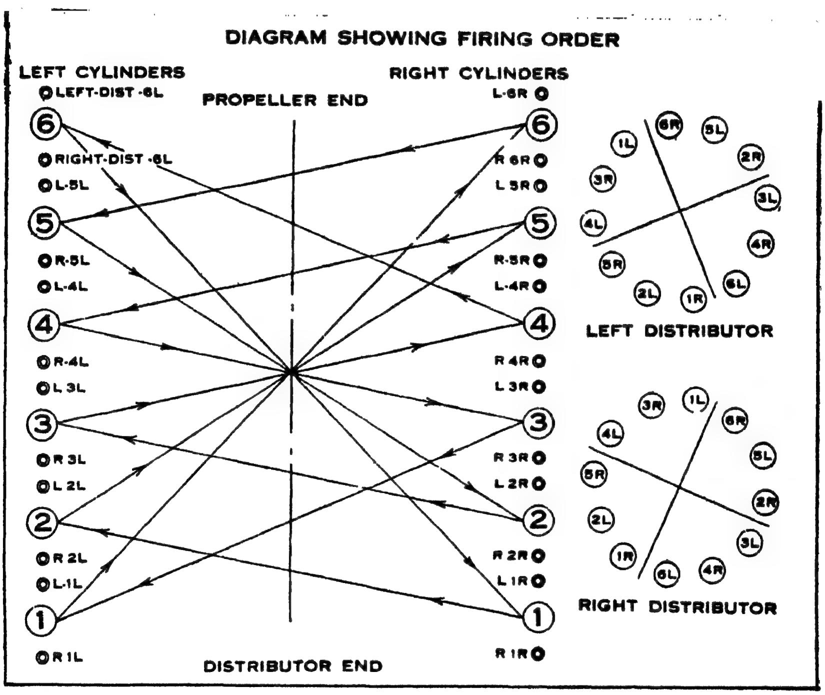
As soon as the engine is started, advance the spark about half way, leaving the throttle at approximately the starting position, and allow the engine to run at idling speed (about 800 r.p.m.) for five to ten minutes or until it is thoroughly warmed up. At the same time test crankcase temperature with your hand. The crankcase should be warm by the time the temperature of the water has increased to 150 degrees Fahrenheit. Accelerate and slow down the engine occasionally to throw the oil up into the cylinders. In extremely cold weather it is possible that the cooling water might warm up more rapidly than the lubricating oil. In this case it would be advisable to stop the engine for a few minutes in order to allow the heat from the cylinders to travel down to the crankcase. In the meantime note the oil gauge pressure. After about three minutes running, at 600 to 800 r.p.m. this should show above five pounds pressure, and at 1,600 r.p.m. up to 30 pounds maximum. Failure to show these pressures may be due to dirt on the relief valve seat. The gauge will show higher pressures when the engine is first started and is cold than after it has thoroughly warmed up. Examine all oil piping for leaks.

Note air pressure gauge. The engine-driven air pump with its regulator, is designed to hold the pressure on the gasoline tank at approximately three pounds. In order to determine whether or not the pump is functioning properly, screw down the pressure regulator adjusting screw. This should cause the pressure in the tank to rise if the pump is operating as it should. Now screw the regulator adjustment up until the pressure is held steadily at three to four pounds. Note water circulation. Temperature gauge should show a steady rise up to not to exceed 200 degrees Fahrenheit. The most efficient temperature will vary with the weather conditions, but will average about 180 degrees Fahrenheit.

Note ammeter reading. At idling speeds the ammeter needle will stand on the "Discharge" side of zero. At about 650 r.p.m., with both switches "on," the needle will stand at zero and at high speeds it should stand on the "Charge" side of zero. When the engine is well warmed up, the throttle may be opened wide (wheels blocked) and the speed of the engine noted. Tachometer should show 1,550 to 1,600 r.p.m. on the ground. Operation of each ignition head should be tested separately by shutting off first one switch and then the other. The engine should show the same r.p.m. in each case. With the throttle wide open, whether the engine is running on one or both sets will make very little difference in the speed (possibly ten or fifteen r.p.m.). At lower speeds (600 to 800 r.p.m.) the effect will be more apparent.

Before stopping the engine, throttle it down to idling speed for a minute or two, then turn the ignition switches to "off" and at the same time open the throttle wide. Opening the throttle will "choke" the engine and cause it to stop immediately. Allowing the throttle to remain in the idling position may permit an overheated plug or particle of carbon to fire the engine spasmodically for some time after the ignition is cut off. Do not attempt to crank an engine immediately after it has been stopped. An overheated plug or incandescent particle of carbon might cause pre-ignition and a disastrous back kick. Allow it to cool off for a few minutes.

Cold Weather Suggestions.—1. Inspect the engine carefully as previously instructed. 2. Put three gallons of hot lubricating oil into the engine crankcase. Oil should be heated in an open top container set into boiling water. 3. Put a sufficient quantity of hot oil into the oil reservoir so that the reservoir will be about two-thirds full after the three gallons placed in the crankcase have been pumped back into it. 4. Remove the vent plug in the side of the oil pump body so that the hot oil may run in to prime the pump. 5. Fill the cooling system with boiling water. Soft water should be used wherever available. Do not use any anti-freeze preparations except those containing alcohol only. 6. Prime the engine and start at slow speed with the throttle partially closed. 7. Accelerate and slow down the engine occasionally to throw the oil up into the cylinders. Run the engine on the ground until the oil has been thoroughly distributed as indicated by the



• **Fig. 443.—Diagrams Showing Firing Order of Liberty-12 Aviation Engine.**

action of the oil pressure gauge and a uniform temperature of the engine. This period need not be continuous and if possible engines should be alternately run for a few minutes, stopped for five minutes and then restarted. 8. Do not attempt to get off the ground until water temperature is at least 160 degrees Fahrenheit.

If the machine is not to leave the ground at once, the engine should not be allowed to remain stationary for more than ten minutes at a time as it will get cold again. After finishing a test or flight, drain all oil and water before the engine has had an opportunity to cool off. Plug No. 250 and sump cover No. 8,129 should be removed to drain the oil from the engine.

A plug of the same number (No. 250) is provided in the bottom of the water pump for the purpose of draining the water. If the engine is installed as a tractor, this plug will be the lowest point in the cooling system. If the engine is installed as a pusher, the tail of the machine should be raised until the propeller end of the engine is higher than the distributor end in order to allow all of the water to drain off. Sparkplugs should be removed from the engine and kept in a warm place if the engine is to stand idle over-night or for any considerable period.

Liberty Engine Trouble.—The diagnosing of gasoline-engine troubles is largely a matter of experience on account of the fact that a symptom may be due to any one of a number of causes. A correct conclusion can only be arrived at by a process of elimination. The different causes, as set forth in the table which follows, have been arranged in the order in which they most frequently occur. Try one thing at a time, and in the order given in the table. Once the trouble is located the remedy should be obvious. It is essential that any troubles be remedied immediately they are located, otherwise serious damage or the entire failure of the engine may result. If engine cannot be turned over under reasonable pressure; 1. Examine water pump for ice. 2. Examine gears for obstruction.

If engine fails to start, it may be due to any one of the following causes:

1. Lack of gasoline. Examine Tank. Examine Shut-off Cock. Examine Trap. Examine Piping. Examine Hose Connections. Examine Carburetor Float Valve.
2. Primed too heavily. (Rotate engine backwards ten or twelve revolutions to clear cylinders of gas.)
3. Insufficiently primed.
4. Throttle too wide open.
5. Throttle not opened wide enough.
6. Water in carburetor.
7. Battery not up to full strength.
8. Loose connection at battery, switch, distributor.
9. Broken wire.
10. Dirt or moisture on outside or inside of distributor.
11. Wires improperly connected.
12. Ignition incorrectly timed.
13. Air leaks in intake manifold.
14. Valves improperly timed.

If engine stops—1-6-8-9-10 above, or 1. Throttle control loosened up or disconnected. 2. Overheated. If engine misses look for the following: Loss of compression due to valve sticking or valve seat caked with carbon or tappet improperly adjusted. Valve may be warped or valve spring broken. The cylinder of piston scored or rings broken or sticking.

If engine fails to develop power look for insufficient throttle opening, insufficient spark advance, insufficient gasoline supply. This may be due to piping or stop-cock capacity too small. Obstruction in piping. Obstruction in trap. Gasoline tank "air bound"—if gravity or vacuum system. Insufficient pressure—if pressure feed system. Obstruction in one or more jets. Improper carburetor adjustment. One or more cylinders missing fire. Engine overheated. Air leaks in intake manifold. Obstruction in carbu-

retor intake. Water in gasoline. Excessive carbon deposits. Poor gasoline. Weak battery or defective generator or an altitude valve improperly set will also cause loss of power, as will back pressure due to exhaust manifold or pipes of insufficient capacity.

If engine overheats check the following: 1. Insufficient water. 2. Insufficient radiator area or capacity. 3. Water pipes too small. 4. Obstruction in water piping or radiator. 5. Failure of water pump. 6. Insufficient oil. 7. Failure of oiling system. 8. Improper carburetor adjustment. 9. Excessive carbon.

Periodical Inspection.—To insure Liberty engines rendering the maximum service they must be inspected daily or at least after every five hours of flight. It is advisable that these inspections be systematically carried out and that the inspector or squad foreman be provided with a form covering the points set forth in the following instructions. Inspectors should be instructed to rigidly adhere to this form and check the different items off as they are attended to.

1. Feel all bearings—see that they are not overheated.
2. See that propeller hub bolts are tight and properly cotter pinned or wired. It is advisable, after every long flight (five hours) to take the cotter pins or lock wires out of these bolts and draw them up as much as possible. New cotters or lock wires should then be fitted.
3. Check pitch and track of propeller.
4. See that propeller hub is drawn up snugly on shaft.
5. Be sure that propeller hub nuts are securely locked.
6. See that all other visible bolts and nuts are tight and properly locked.
7. Examine all valve springs carefully.
8. Squirt a little light oil through the valve springs onto each valve stem.
9. Examine throttle, spark and altitude adjustment controls. Be sure that they work freely, permit full throw of throttles and distributors that have not become excessively loose.
10. Test all cylinders for compression as previously instructed.
11. Try rocker arms—they should all be free when the valves which they operate are seated.
12. Check tappet gap of all valves—piston at firing point.
13. Check valve timing with timing disc.
14. Examine radiator, water piping, pump, water jackets and all connections for leaks of the cooling system.
15. Fill cooling system. Note—If temperature is below freezing, follow cold weather instructions.
16. Examine tanks, trap, piping and all connections for leaks.
17. Drain water trap.
18. Fill gasoline tank.
19. Drain oiling system by taking out plug in bottom cover of pump body.
20. Remove rear pump cover plate which will release oil pump screen.
21. Clean screen thoroughly with a brush and gasoline.
22. Replace screen and cover plate using a new gasket if the old one was damaged in removing cover.
23. Examine reservoir, cooler and all piping and connections for leaks.
24. Oil thrust bearing.

25. Replace all hose connections, either for water, oil or gasoline, which show any signs of deterioration.
26. Examine all electrical connections at generator, regulator, switch, battery and distributor to see that they are clean and tight.
27. Examine all wiring to see that insulation has not become abraded and that no wires are in contact with metal parts of the engine.
28. Clean distributors as instructed.
29. Oil generator and tachometer drive.
30. Examine plugs for cracked or loose porcelains. This should preferably be done immediately after the engine is stopped and while the plugs are hot.
31. Check contact breaker clearance and examine contact points.
32. Check timing of ignition and synchronization of distributors.

Caution:—Leave the ignition switches and the gasoline shut-off cock in the "Off" position. If the switches are left "on," the battery will discharge through the ignition system and generator and it will be necessary to either recharge or replace it before the engine can be started again. With both switches turned off, the ammeter needle should stand approximately at zero.

Overhaul and Repair.—After periods ranging from 100 to 150 hours run, every Liberty engine should undergo a thorough inspection and in order that this may be done properly it should be taken out of the plane and completely disassembled.

U. S. A. Standardized engines are made up of a combination of units or assemblies and the units, in turn, of a number of sub-assemblies. In the instructions which follow, the various units are described in the order in which they should be dismounted. It will be found that time and space can be saved and confusion of parts avoided if each assembly be disassembled, inspected and overhauled as it is dismounted. The sub-assembly should then be assembled and laid to one side until such time as the whole engine is ready to reassemble. The combining of the sub-assemblies into a complete engine is outlined in proper sequence.

Shorter and better methods of handling the work may be developed, but, in the main, the plan outlined here will be found most satisfactory. A bench or other suitable place on which parts and tools may be laid out should be provided. The mechanic should form the habit of laying out his tools in a definite order so that the one desired may be readily reached. Have a bucket of kerosene handy in which parts may be washed. Waste should not be used for wiping parts. Threads or lint are likely to stick to the surfaces and eventually find their way into oil passages. Pieces of clean cloth or rags are preferable. Nuts such as those used to hold the cylinders down on the crankcase should be screwed on to their proper studs after disassembling—not only to protect the studs but to avoid confusion in reassembling. Cotter pins and lock wires which have been badly bent should not be used again, as a straightened wire or pin is apt to break in service.

Great care should be taken to prevent pieces of cotter pins, lock wire, chips or any small parts from falling into the crankcase or any part of the engine. They might work into the gears or oil passages and cause consider-

able damage. Have an oil can, full of clean oil, handy at all times. Oil all bearing surfaces before assembling. Oil all parts which are press or drive fit. Oil all bolt and stud threads. Exhaust manifold studs should be greased with a graphite paste.

Each step in the process of assembling should be finished as the work progresses. Do not leave a bolt loose or a nut not cottered with the idea of coming back to it later on. Never "slack off" or loosen a nut in order to line up the notch in the nut with the cotter pin hole in the bolt or stud. If it cannot be tightened still further with reasonable effort, in order to bring the next notch in line, replace it with another nut. All sub-assemblies, as they are overhauled and put together, should be covered to protect them from dust and dirt. Remember that failure to properly attend to the smallest detail of inspection or assembling may result in the failure of the engine and the plane which carries it and perhaps the loss of one or more human lives, not to mention the loss of a valuable airplane.

The following tools are necessary: Canvas tool roll. Piston pin drift pin. Handle for T shaped socket wrenches—ten inches. Handle for T shaped socket wrenches—twelve inches. Handle for T shaped socket wrenches—eight inches. Socket wrench combination for 5/16 and 1/2 inch. Crankcase lower half stud nut wrench. Double end open wrench—3/8 and 7/16 inch. Double end open wrench—1/4 and 5/16 inch. Carburetor wrench. Cold chisel. Cotter pin puller. Distributor wrench. Combination pliers. Bicycle wrench. Six-inch pliers, side-cutting. Hammer—one pound. Adjustable spanner wrench. Socket wrench—T shaped, 3/8 inch assembly. Sparkplug wrench. Raw-hide hammer. Valve spring compressor assembly. Eight-inch adjustable open end wrench—one inch open. Thickness gauge, 3 5/8 inch (.002 inch to .015 inch). Oil and grease gun. Valve grinding tool. Valve grinding compound. Scraper. Screwdriver—large (fourteen-inch square shank). Six-inch mill file. Drift pin. Double end socket wrench, L shaped—3/8 inch. Double end socket wrench, L shaped—5/16 inch. Double end socket wrench, L shaped—1/4 inch. Double end socket wrench, L shaped—7/16 inch. Double open end wrench for camshaft driving shaft and generator driving shaft nuts. Propeller hub lock nut wrench. Propeller hub retaining nut wrench. Screwdriver—small (six inches). Timing disc. Timing disc pointer.

Electrical Equipment.—The ignition system used on Liberty 12 aero engines is known as the Generator-Battery type. The system comprises two independent breaker and distributor mechanisms or heads, identical in every respect and each one firing all twelve cylinders. These distributors are supplied with electrical energy from two sources. For starting and for idling speed up to 650 r.p.m., current is drawn from a specially constructed four cell or eight volt storage battery. The battery is very light and carries very little liquid or electrolyte (barely enough to fill a hydrometer syringe besides what is absorbed by the plates and separators). Nevertheless it has sufficient capacity to ignite the engine at full speed for three hours. It is so constructed that, even though it be turned upside down, it will still continue to function properly.

Generator.—In addition to the battery, a positively driven generator is provided, so geared that it runs at one and one-half times crankshaft speed. As stated above, electrical energy for starting and idling speeds is supplied by the battery. As the engine speed is increased, the generator "builds up" and its output grows greater until, at about 650 r.p.m., the generator voltage equals that of the battery. The maximum generator output exceeds the requirements for ignition so that, at speeds above 650 r.p.m. the direction of flow of current is reversed and the excess output of the generator goes to recharge the battery. The rate at which the battery will be recharged will depend upon the condition of the battery. With an almost discharged

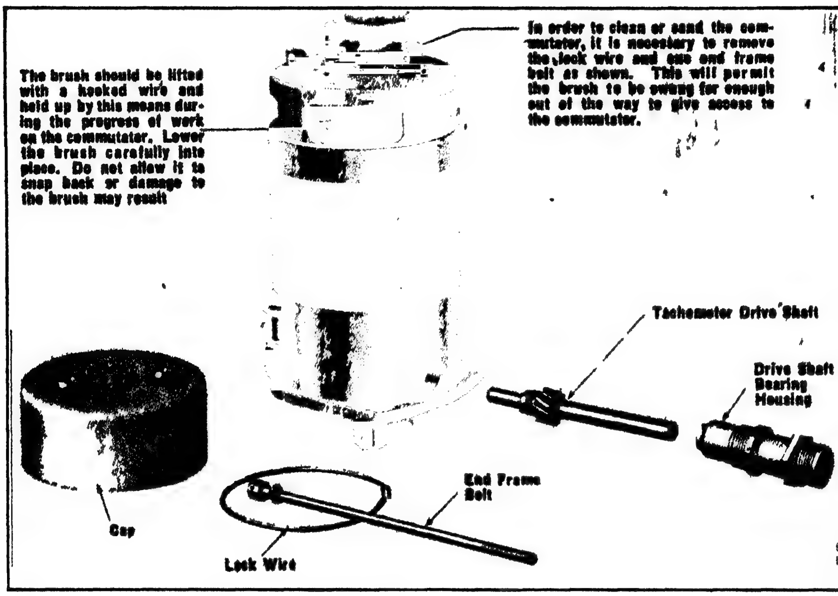


Fig. 444.—The Liberty Engine Generator Removed from the Motor.

battery the rate will be about ten amperes but will diminish as the battery voltage rises until the battery is completely charged, when the charging rate will be just sufficient to maintain it in a properly charged condition. The generator is shown at Fig. 444 removed from the engine and with brush cover taken off to expose the commutator.

The generator is controlled by a "voltage regulator" which prevents the output exceeding a predetermined figure. In view of this fact, the generator will supply current for ignition indefinitely, without the battery, so long as the engine speed is not allowed to drop below 500 r.p.m. It is not possible to crank the engine fast enough to start it on the generator, however.

Switch.—A duplex ignition switch is provided which will permit either one or both distributors being turned "on." This switch, which is shown at Fig. 445, is so constructed that either set of ignition alone can be used without connecting in the generator. In starting, only one side should be used as, with both switches "on," the generator is connected to the battery. Under these conditions the discharge from the battery through

the generator before the engine is started would be an excessive drain on the battery. It is essential that both switches be "on" at all flying speeds, however. The ignition switch has an ammeter incorporated in it and this ammeter should be watched occasionally as it indicates the amount of current flowing to or from the storage battery. If the ammeter shows a discharge at any speed above 650 to 700 r.p.m. with both switches "on," it is an indication that something is wrong with the generator circuit, and that all electrical energy is being supplied by the storage battery. If the ammeter stands at zero under the same conditions it indicates that the storage battery is not receiving a charge, but that the ignition is being carried by the generator.

Distributors.—To return to the distributors—the circuit breaker mechanism for each head is identical with that used in any high grade magneto with two exceptions, as follows: Two main circuit breakers, connected in parallel, are provided instead of one. The two breakers are timed to operate simultaneously and are provided in duplicate as a precautionary measure. An auxiliary circuit breaker, the function of which is to prevent the production of a spark when the engine is turned backward or "rocked" is also provided. This auxiliary breaker is connected in parallel with the other two through a resistance unit which reduces the amount of current flowing through it. The breaker is so timed that it opens slightly before the other two when the engine is turned in a forward direction. The opening of the main breakers then results in the production of a spark. When the engine is turned in a backward direction the two main breakers open first and no spark is produced due to the fact that the current continues to flow through the coil through the auxiliary breaker but in diminished quantity due to the resistance unit. By the time the circuit is opened at the auxiliary breaker the intensity of the magnetic field of the coil has weakened to such an extent that no spark is produced. A transformer coil is incorporated in the Bakelite cover of each distributor head as shown at Fig. 442.

Advantages.—The advantages which this type of ignition is said to present over the magneto system are: 1. Easy starting—a spark of greater intensity is produced at cranking speed than at flying speed. 2. Reliability—two distinct distributor mechanisms, each one igniting all twelve cylinders through separate sparkplugs, each distributor head being fitted with two sets of breaker arms and contact points. Two distinct sources of electrical energy—battery and generator. 3. Safety—the auxiliary breakers prevent the possibility of a "back kick." 4. Great range of spark timing control—a spark of the same intensity is produced whether advanced or retarded. 5. Ability of the pilot to determine whether the electrical equipment is functioning properly through the medium of the ammeter. 6. Simplicity—distributor heads are driven direct from the camshafts without the use of gears and extra shafts. 7. Long life—the distributor heads run at slow speed (one-half crankshaft speed) hence the wear will be slight. 8. As the distributor and breaker are advanced and retarded together, they are always properly timed with relation to one another. Consequently there is no possibility of pre-ignition due to the high tension current being carried to the wrong plug. 9. The spark is hot and of short duration so that no "track" trouble is experienced. With the magneto the high tension

impulse tapers off gradually and a spark is drawn out by the rotor brush *after it has left the distributor segment*

High Tension Wiring.—*Disconnect all high tension cables from the* sparkplugs Remove the screws which hold the cable tube clips in place Release four clips and unscrew two composition nuts on each distributor head The distributor head, cables and the cable tube may now be lifted

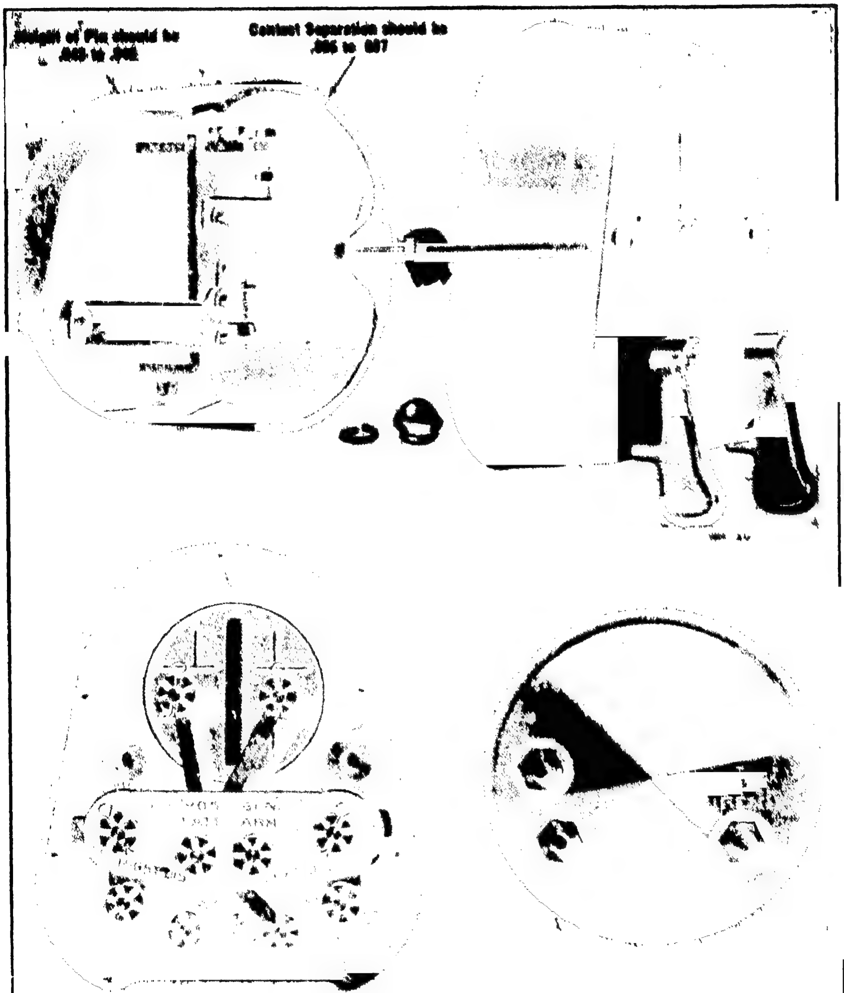


Fig. 445.—View Showing Construction of the Switch Used in the Liberty Motor Ignition System.

off together They should be handled carefully to prevent any possibility of breaking the distributor covers and damage to the cables.

Examine these cables carefully, especially at any points where they may be bent sharply or where they may come in contact with any metal parts. See that the insulation is intact and that the terminals are firmly attached.

Examine inside surface of distributor cover and particularly the contact segments and the path of the rotor brush. The surface should be smooth and free from scores or scratches. Wipe out any carbon dust carefully with a soft cloth moistened with alcohol or gasoline.

Examine the distributor shaft for lost motion between the distributor driving flange and the driver. This should not be more than enough to allow about 1/16 inch motion at the end of the rotor arm. Distributor base:—Disconnect the cross reach between the two distributor assemblies and take out bolts from the distributor base flanges. This will permit the distributor base assemblies to be taken off.

Check the gap of each breaker with a thickness gauge. This can best be done with the breaker block on the wide lobe of the cam. The gap for all three breakers should be from .010 inch to .013 inch. Examine the condition of the contact points. These points should be bright and in case they have become pitted to any extent, they can be smoothed down on an oil stone. They are very hard and it will be impossible to file them. Examine the rubber buffers against which the breaker arm springs bear. These rubber buffers are vulcanized to the breaker arm and in case they have deteriorated to any extent due to contact with oil, the whole breaker arm assembly should be replaced. Examine the breaker arm springs. Be sure they have not begun to crack around the slotted hole through which they are bolted to the bus bar. The tension of the spring on the auxiliary or middle breaker should be from sixteen to twenty ounces when the contact is open. The tension on the main breaker arm springs should be from 28 to 30 ounces, when the contact is open.

Examine the resistance unit through which the current passes to the middle breaker. The coils of this resistance unit should be separated throughout the length of the unit. If the coils of resistance wire come in contact with one another, the total resistance of the unit will be decreased and the proper functioning of the middle breaker will be impaired. Examine the condition of the face of the cam. This should not show excessive wear and should be nicely burnished. Examine the condition of the fiber blocks which bear against the cam surface. The fiber should extend approximately 3/62 inch beyond the metal and should show a smooth bearing surface.

Look at the carbon brush in the end of the rotor arm. If the brush has worn down to such an extent that it is less than 1/4 inch long, a new brush should be fitted. In putting in the new brush and spring, press the brush into place in the rotor arm with a small punch as far as possible. This will properly seat the spring and will prevent the brush from extending too far out of the guide. The rotor brush is a special composition and requires no lubrication.

Examine the ball-bearings which carry the distributor camshaft for radial and lateral lost motion. This should be only barely perceptible. If it is deemed advisable to replace one or both of these bearings, the rotor arm should be removed by taking out the screw D (see Fig. 442). The shaft may now be driven out through the cam. This should be done with a small punch which will permit the key to remain in the cam. In reassem-

bling the distributor the bearings should be packed with vaseline and the felt washer with which each cam is filled should be thoroughly saturated with a good light oil. This oil will work out through the small holes drilled in the face of the cam and will properly lubricate the contact arm blocks. *It is advisable also to put a little vaseline or thin grease on the outside surface of the cam. The studs on which the contact arms are mounted* should also be oiled. Be sure that the contact points are properly adjusted and that the lock nuts are drawn up snugly.

The transformer coil is built into the distributor housing cover. All connections are made inside and the whole is covered by a fiber plate which is sealed in place. It is not advisable to remove this cover plate for any purpose whatsoever. In replacing the rotor arm be sure that the drive pin in the cam properly enters the hole in the arm. Information as to timing and adjusting the distributor assemblies will be found under the heading of timing.

To dismount the generator remove the four nuts which will permit the generator to be lifted off of the engine base. Examine the condition of the splines or keys on the lower end of the shaft. These should not show undue wear. The tachometer drive assembly complete can be removed by screwing out the bearing assembly. (See Fig. 444.) This will permit the worm gear shaft to be withdrawn. The worm gear and the bronze bushings in which the shaft runs should now be examined as to their condition. They should be free from bad cuts or scores. Before inserting the shaft again in the bearing, the recess between the two bushings should be filled with vaseline or other light grease. Remove the cap from the upper end of the generator by taking off the two terminal nuts.

Examine the condition of the commutator and brushes. In case the commutator is burned or rough it should be polished with a very fine piece of sandpaper. The best possible condition of the commutator is shown when it takes the form of a bluish polish, which should not be mistaken for a burned commutator. When the commutator carries a blue polish on it, it should be allowed to remain this way and only receive an occasional wiping off with a soft rag.

In case the commutator is badly cut or scored, it should be turned down in a lathe a sufficient amount to smooth it up. The mica should then be "undercut." This work should only be done by experienced mechanics. If the commutator was found to be scored, the brushes will also be in bad condition. These brushes are soldered to the brush holders and in case of a replacement a new brush holder and brush should be fitted. Brushes that have been roughed up or new brushes should be sanded to fit on the commutator by wrapping a strip of very fine sandpaper at least half way around the commutator and drawing through under one brush at a time. This will form the brush to the curvature of the commutator. It is essential that the brushes properly fit the commutator and the work should only be done by an experienced mechanic.

Examine the wire leading from the field coils to the generator terminal and its connection with this terminal. Any dirt or excess oil should be wiped off before replacing the cap. The ball-bearing on the upper end of the generator shaft can be lubricated through the oil cup in the cap. This

bearing should receive a few drops of oil after every long flight, but should not be lubricated excessively as the oil is likely to run down over the commutator.

Voltage Regulator.—This device consists of an iron core on which are wound three coils, the connections of which are shown on the circuit diagram, Fig. 446. The frame of the regulator carries a pivoted armature fitted with adjustable contact points at one end. The contact points are normally held together by an adjustable spring.

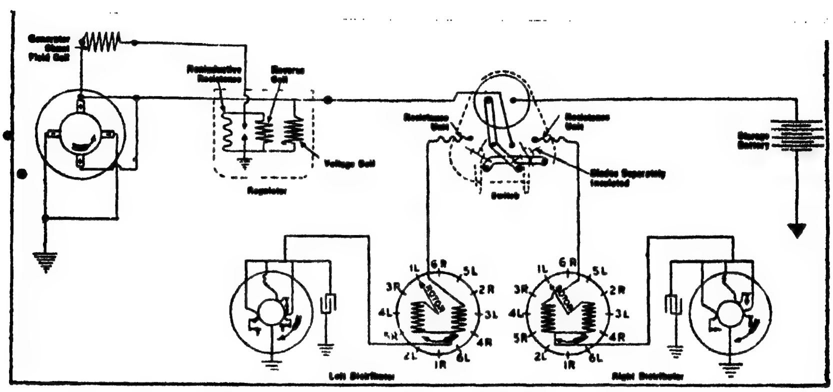


Fig. 446.—Circuit Diagram for Liberty Motor Ignition System.

The function of the voltage regulator is to prevent the generator from delivering more than a pre-determined supply of electricity. The regulator will need practically no attention with the exception of an occasional checking up of the contact separation and the length of the brass pin set into the opposite end of the armature. This pin should extend from .043 inch to .045 inch above the surface of the armature. When the armature is pressed down so that this pin bears against the end of the core the gap between the contact points should be from .005 inch to .007 inch. If the contacts are burned or pitted they may be smoothed down with an oil stone. The tension of the adjusting spring should not be altered except by an expert and with the aid of proper instruments. See that all terminals are clean and tight.

Ignition Switch.—The ignition switch should be inspected each time the engine is taken out of the plane for overhauling. Two resistance units are mounted on the back of the ignition switch. These units should be examined to see that the coils do not come in contact with one another. If this should happen the total resistance of each unit would be reduced, which would result in the burning of the distributor contact points. It would also affect the proper functioning of the auxiliary contact arm in the distributor. If these resistance units should burn out entirely, it would shut off the ignition.

The three screws on the face of the switch should be removed and the switch contacts under the cover examined to see that they have the proper tension and that they are clean. The ammeter which is part of the switch

assembly should require no attention. In case the pointer sticks the instrument should be replaced, as it is very important that the ammeter should indicate accurately, so that the exact condition of the generator, voltage regulator and the ignition can be determined. It is advisable that the replacing of a resistance unit or an ammeter be done only by electricians familiar with this work.

Preparing Battery for Service.—One storage battery (bone dry) was packed in each engine crate. To prepare this battery for service: 1. Remove vent plugs. 2. With syringe provided in crate, fill each cell to a depth of one inch over bafflle plate with electrolyte of 1.255 specific gravity. If electrolyte of 1.255 specific gravity is not at hand, it can be made by mixing chemically pure sulphuric acid of 1.835 specific gravity (66 degrees

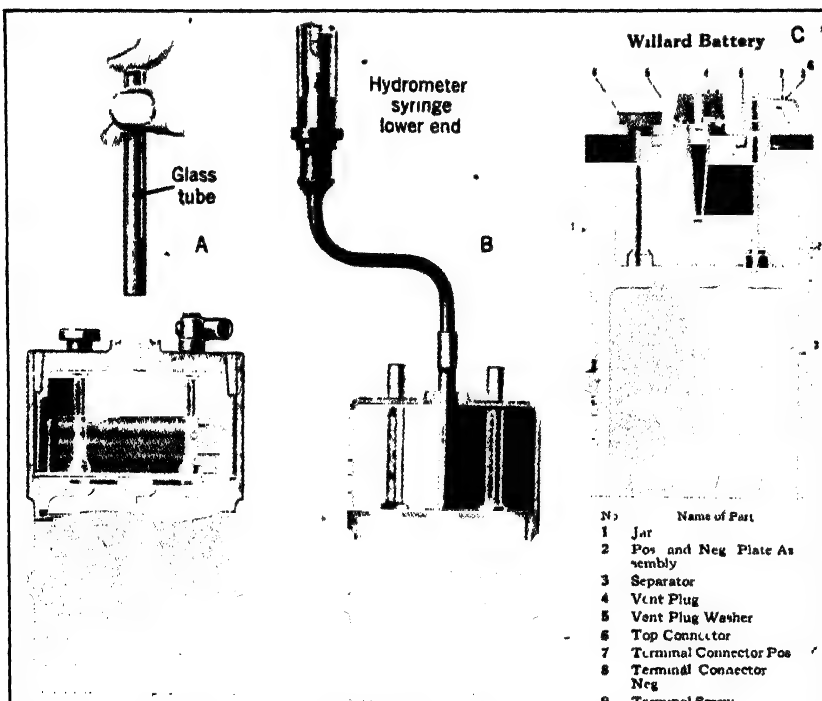


Fig. 447.—Illustration Showing Method of Putting Storage Batteries Used in Liberty Aviation Engine Ignition System in Service.

Baumé) and pure water in the proportion of one part acid to three parts water by volume. When mixing use nonmetallic vessels and pour the acid into the water in a thin stream. Never pour the water into acid. Allow the mixture to cool before reading the specific gravity and using. 3. After fifteen minutes, check to see that electrolyte is one inch above baffle plate. If electrolyte is not one inch above baffle plate, additional electrolyte should be added. 4. Allow battery to cool for eight hours. 5. Charge battery at seven-tenths (.7) amperes for 70 hours. 6. Take a test discharge of twenty amperes from battery for fifteen seconds, each cell voltage

to be taken immediately. If each cell is 1.55 volts or over (at 80 degrees Fahrenheit) with the above current flowing, battery is O. K. 7. Allow the battery to stand fifteen minutes, then with syringe mentioned above remove surplus electrolyte from each cell down to baffle plate, battery being in upright position. Replace plugs, screwing down firmly with fingers. The battery is now ready for service.

Flushing.—Battery should be flushed once a week, as follows: Vent plug should be removed and each cell filled with distilled water to a height of one inch above the plate as shown in Fig. 447 A, allowed to stand for a minimum of two minutes (not over five minutes) and surplus water taken off down to plate with hydrometer syringe, as shown at Fig. 447 B.

Charging and Adjusting Gravity.—Specific gravity can be adjusted in the following manner: 1. Battery should be flushed, as described above. 2. If battery is in a discharged condition, it should be put on charge at one ampere and charged until the terminal voltage with this current flowing has risen to a maximum, i.e., shows no further rise for a period of one hour. (Vent plug should be left out during this charge.) The battery should then be tipped on its side and gravity taken with hydrometer provided with a bent hard rubber tube. Be sure to return electrolyte to the cell from which it was taken.

In batteries fully charged, gravity should be between 1,290 and 1,310. If gravity is not correct on taking gravity readings, battery should be held upside down for five or six minutes and electrolyte allowed to run into a rubber or glass jar. The electrolyte removed should be adjusted to 1,300 specific gravity by the addition of distilled water, or 1,400 acid, as the case may be; then each cell should be filled with this electrolyte until the level is one inch above plate. Battery should then be allowed to stand a minimum of five minutes (not more than ten minutes) and surplus electrolyte removed to top of plate with hydrometer syringe, battery in upright position. Replace vent plugs. The aviation battery is shown in cross-section in Fig. 447 C, which shows Willard type SY-13. In this cut the part number of each separate part and a key list of material giving the name of each part is also given.

Water Outlet Headers.—Remove the nuts which hold the water outlet headers in place, and lift these headers with the extension tube and the two hose connections off. The clamps on these hose connections should be taken off and the hose examined. If the hose connections show any signs of deterioration they should be replaced before reassembling. Lay the outlet header gaskets to one side where they will not be damaged and if damaged in any way they must be replaced when parts are reassembled.

Intake Headers.—Disconnect the water inlet elbows by removing the cap screws. Remove the intake header nuts and the washers. This will permit the intake headers to be dismounted. Examine the gaskets and see that they do not overlap the openings in the elbows or the holes in the intake header. If the gaskets appear to be in good condition, put them back on the studs in their former position and screw on the nuts to hold them in place. Leave the water connection elbows and hose connections attached to the cylinders.

Camshaft Housing Units.—Each of the two assemblies consists of a camshaft with its bearings and gear—the rocker levers—a camshaft housing with its covers and the camshaft drive shaft with its gear, bearings, and housings. The assembly also included all bolts, nuts, cotter pins and small parts shown at Fig. 448. All parts of the two assemblies (right and left) are identical and interchangeable with the exception of the camshafts themselves and the camshaft housing covers. Each shaft is stamped with a serial number on the soft plug in the end opposite the flanged end. Right-hand shafts bear the letter R and left-hand shafts bear the letter L. The housing covers are machined in place on the housing and consequently will not interchange.

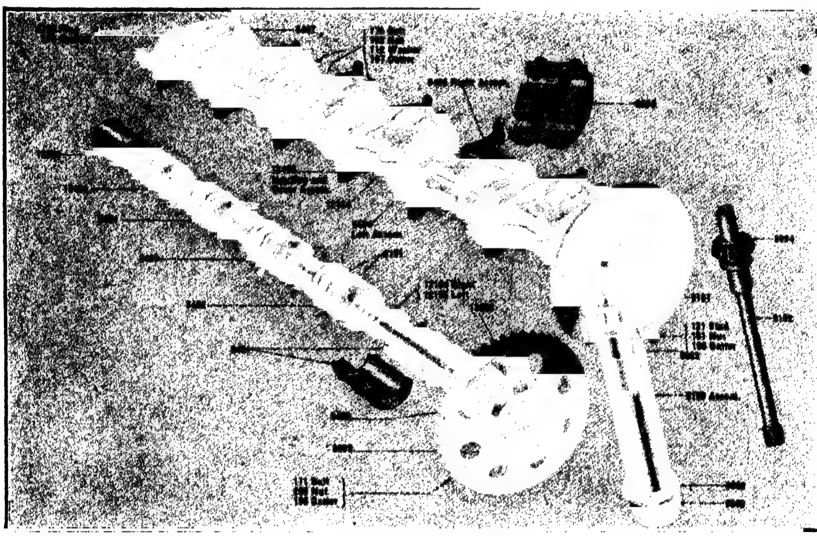


Fig. 448.—The Overhead Valve Operating Gear of the Liberty Aviation Engine.

Disassembling and Inspection.—The following instructions apply to each of them. The housing assemblies should be dismounted from the engine in the following manner:

1. Disconnect oil pipe by unscrewing nut. The connection No. 8,121 may then be slipped off. Put the two gaskets back in place and screw on the nut No. 8,122.
2. Unscrew the packing nut and slide it and the felt packing washer up on the housing. If the felt washer does not fit sufficiently tight to hold both in place, tie them so that they will not slip off.
3. Remove twelve nuts and washers and lay them to one side in the order in which they were originally assembled.
4. The unit may now be lifted off the engine. Raise it carefully and evenly so as to avoid risk of bending the studs.
5. Put the washers and the nuts back in place on their proper studs which project from the top of the cylinders.
6. Examine all rocker lever shafts with respect to the fit in bearings.

These shafts should have from .005 inch to .010 inch end play and from .001 inch to .0015 inch clearance in bearings. The bearing fit should be "free running" and the radial clearance or shake should be barely perceptible.

7. Remove camshaft housing cover bolts No. 178, washers No. 112 and covers No. 8,095 and lay them to one side in the order in which they were originally assembled.
   Caution—The joints between the covers and housing are "lapped" and the covers are marked to fit in their proper place. Care must be exercised in handling these parts so as to avoid marring the surfaces which come in contact.
8. Examine rocker levers. As each tappet has been adjusted as to clearance for the particular valve which it operates, the rocker levers should be laid to one side in the order in which they were originally assembled.
9. Examine the tappets to see that they are tight and that the faces do not show undue wear.
10. Examine the rollers to see that they are not cracked. See that the rollers are free running and show no flat spots. The bearing for the rollers is a steel thimble which is a free fit in the roller. This thimble is pinched tight in the rocker lever fork by the pin which is riveted at assembly in rocker lever. The permissible shake is .001 inch. Roller should have .010 inch side play in the fork of the rocker lever.
11. Examine rocker lever bearings. See that they are clean and show no evidences of "scoring." Scored shafts should be smoothed up with a fine file and sand paper. Bearings should be touched up with a scraper if scratched.
12. Remove camshaft. Take out seven camshaft bearing lock screws with their gaskets. This will permit the camshaft and six bearings to be withdrawn through the gear end of the housing. The rear bearing may be pushed out through the other end of the housing. The camshaft bearings are so fitted that they should come out easily. Should they stick the four screws No. 8,219 and bearing cover plate No. 8,376 may be removed and the shaft driven out with a soft drift.
13. Examine camshaft bearings. These bearings should have a clearance on the shaft of from .001 inch to .003 inch, which means that the "shake" will be just perceptible with an oil film in the bearing. The gear end bearing should have an end play of from .001 inch to .003 inch between camshaft flanges. All bearings should be clean and free from "scores." The sets of camshaft bearings are "stepped." That is, the bearing at the gear end has the greatest outside diameter. The next one is $\frac{1}{32}$ inch smaller; the third bearing is $\frac{1}{32}$ inch smaller than the second, etc. Therefore to facilitate reassembling, lay the bearings out in the order in which they were disassembled and be sure they are replaced in proper positions.
14. Examine camshaft. See that all bearing surfaces are bright and smooth. Inspect cams carefully for cracks and soft spots.
15. Examine camshaft gear. See that the teeth are all intact and do not show undue wear. Unless the gear is defective it should not be

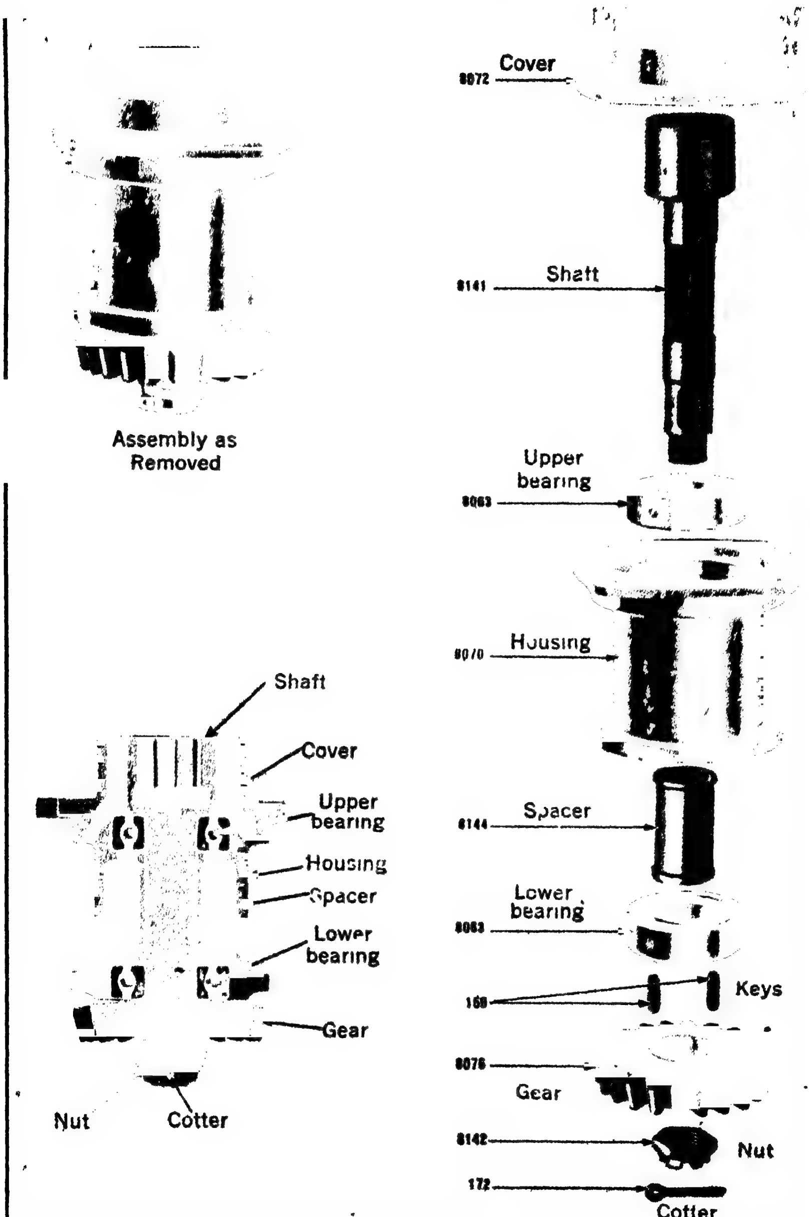


Fig. 449.—Camshaft Lower Drivegear Assembly of the Liberty Engine and its Parts.

necessary to remove it from the shaft.

16. Remove camshaft upper drive shaft. Take off nuts No. 101 and washers No. 111, which will permit the removal of housing No. 8,109. A gasket, No. 8,139, $\frac{1}{64}$ inch thick, will be found between the flange of No. 8,109 and the camshaft housing. (See Fig. 448.)
17. Withdraw the drive shaft and examine the bearing surfaces and the bushings in which they run. See that the gear is tight on the shaft and that neither the gear nor the splines at the lower end of the shaft show undue wear. Should it be necessary to change either of the camshaft drive shaft bearings, the new bushings should be oiled on the outside and pressed in carefully under an arbor press or in a vise. They should then be reamed to size—the upper one, No. 8,107, to .875 $\pm$ .0005 inch and the lower one, No. 8,092, to 1.125 $\pm$ .0005 inch. All parts should be washed in gasoline or kerosene and blown off with an air jet. Especial care should be taken that all oil passages are clear and free from dirt.

Assembling.—1. Fit the camshafts with their bearings in the camshaft housings. If it has been found necessary to replace any of the bearings, the new ones should be tried in their proper places in the housings to see that they are what is termed a "wringing" fit. That is, they should be free enough so that they may be turned with a drift to line up the lock screw hole and still there should be no "shake" perceptible. If a new shaft is to be fitted, all bearings should be tried in their proper places on the shaft. The diametrical clearances should be from .001 inch to .003 inch. Oil all bearing surfaces with clean oil before assembling. Be sure that punched holes in gasket No. 8,379 and end plate No. 8,376 line up with oil hole in bearing No. 12,250, and that screws are wired. Line up the lock screw hole carefully and put the screws in place with their gaskets. If a new shaft or new gear is to be fitted be sure that camshaft gear bolts No. 171 are in place before putting on bearing No. 8,111, No. 8,112. In the above case leave the gear off until later on in the process of assembling the engine.

2. Put the rocker levers and housing covers in place and test each lever for free operation.

3. Screw on the cover No. 8,088 against its gasket No. 169.

4. Put the camshaft drive shaft (upper) and its housing in place with the gasket No. 8,139 between the flanges. The shaft should turn freely after the housing bolts are drawn up and the end play should not exceed .008 inch. If the original gears are used, they should be meshed so that the markers come in line. Permissible "back lash" in these gears is .005 inch to .010 inch.

Lower Camshaft Drive Shafts.—These assemblies, No. 8,208, are illustrated at Fig. 449 and each embodies the parts clearly shown in sectional view. Take off four nuts No. 101 which will release the cover No. 8,072, and permit the assembly to be lifted out. Examine gear and splined socket. Try end play in bearings. It should not exceed .004 inch. To disassemble, remove nut No. 8,142 and draw off gear with gear puller. **Caution—Do not attempt to drive shaft out through gear.**

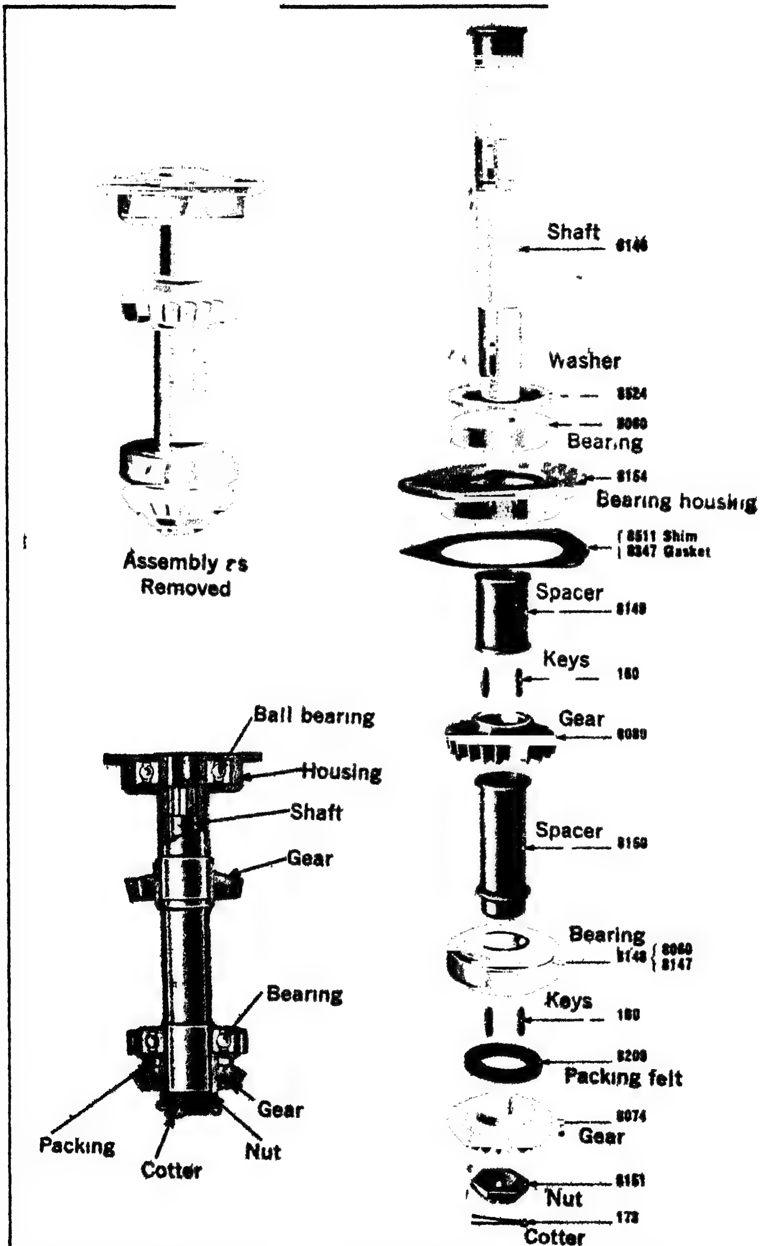


Fig. 450.—Generator Driveshaft Assembly of the Liberty Engine and its Parts.

The keys in shaft will not pass through the lower ball-bearing. After drawing off the gear the keys may be lifted out and the shaft may then be tapped or pressed out through the bearings and spacer. The parts of this unit should be assembled in the reverse order to the above, care being taken that the keys are properly seated in the shaft and that all parts are drawn up snugly by means of the nut No. 8,142. **Note—Unless some part of this assembly is defective, it is not advisable or necessary to take it apart.**

Generator Driving Shaft Assembly.—Remove a flat head screw. This will permit the generator drive shaft assembly, as shown at Fig. 450, to be lifted out. One or more metal shims .002 inch thick will be found between the bearing container No. 8,154 and the crankcase. The number of shims will be sufficient to insure the proper mesh of the lower generator drive shaft with the crankshaft gear. They should be tied together and hung up or laid to one side so that they will not be lost or damaged. Examine gears for wear. End play in bearings should not exceed .004 inch. Remove nut No. 8,151. Do not attempt to drive shaft out but draw off gear No. 8,074 with a gear puller. The same construction is employed here as in the lower camshaft drive shaft assembly and the same caution applies. Remove keys No. 160 and draw off spacer and bearing assembly No. 8,148. Remove gear No. 8,080 and keys No. 160, which will permit the spacer No. 8,149 and bearing No. 8,060 to be taken off.

The bearing No. 8,060 is held in place in the container No. 8,154 by means of the lock ring No. 8,524. If it should be found necessary to remove this bearing, the lock ring may be compressed with a pair of round nosed pliers. The parts of this unit should be assembled in the reverse order to the above, care being taken that the keys are properly seated in the shaft, and that all parts are drawn up snugly by means of the nut No. 8,151, shown at Fig. 450. **Note—Unless some part of this assembly is defective it is not advisable to take it apart as it has been carefully inspected as to proper alignment of gear teeth.**

Liberty 12 Oiling System.—Oil supply is carried in a reservoir provided with a suitable means for cooling it. Oil is led from this reservoir to the connection on the right side of the oil pump body marked "Oil In." It is filtered at this point through a large area fine mesh screen. A delivery pump of the gear type takes the oil up after it has passed through the screen and delivers it under considerable pressure to a distributor pipe running the entire length of the crankcase. Opening out of the passage between the pump and the distributor pipe is a pressure regulating valve designed to maintain a pressure not to exceed 50 pounds per square inch on the oiling system. Pipes are fitted in the case leading from the distributor pipe to the main crankshaft bushings.

The crankshaft is hollow and in the center of each main bearing a radial hole is drilled through the shaft into the hollow center. This hole in the shaft registers with the corresponding hole in the bearing bushing once every revolution of the shaft, at which time a small quantity of oil is forced through into the hollow crankshaft. A passage leads from each hollow main bearing to the adjacent crankpin, which is also hollow. A radial hole is also drilled through each crankpin and carries the oil out on the surface of the pin. Oil grooves and passages in the connecting rod bushings insure

proper lubrication for both the forked and plain connecting rods.

The excess oil thrown off the rapidly moving connecting rod ends forms a mist which lubricates the piston pins and the cylinder walls. Part of the oil conducted to the main crankshaft bearing at the propeller end of the engine goes through a passage around this bearing and up through pipes to the propeller end of the camshaft housings. From the end of the camshaft housings it is led around the end camshaft bearing to a passage drilled diametrically through the bearing midway of its length. Once every revolution of the camshaft a hole drilled through the camshaft into its hollow center registers with the oil passage through the bearing. Thus once every revolution a small quantity of oil is forced into the hollow camshaft.

The oil is led through the camshaft and out through holes drilled in it to each camshaft bearing. The excess works out of the ends of these bearings and collects in small reservoirs to a depth of about 1/4 inch. The cams, in revolving, dip into this oil and splash it over the cam rollers and into the pockets in the rocker lever shafts. From these pockets it is led through the hollow rocker shafts to the rocker shaft bearings. The excess oil eventually finds its way to the gear end of the camshaft housings, over the gears and down the drive shaft housing into a chamber just above the oil pump. The excess oil thrown off in the crankcase by the connecting rods collects in this same chamber when the engine is inclined so that the propeller end is high. If the propeller end of the engine is low, this oil collects in a small sump or chamber at the propeller end of the crankcase. Immediately above the oil delivery pump is located an oil return pump consisting of three gears, and driven by the same shaft as the delivery pump. The function of this oil return pump is to draw the excess oil out of the crankcase and return it to the oil reservoir. One-half of this pump draws oil from the sump at the propeller end of the crankcase and the other half draws oil from the sump at the distributor end of the crankcase. Both halves of the pump deliver oil to the connection on the left side of the oil pump body marked "Oil Out," from which point it returns to the oil reservoir.

Oil Pump.—The oil pump assembly No. 8,200 embodies the parts clearly shown in illustration at Fig. 451. Dismount the oil pump assembly from the engine by removing nuts and washers. Then:—

1. Take out the bolts No. 116, nuts No. 101 and washers No. 111. This will permit the cover No. 8,381 of the oil pump body to be taken off and will release the oil strainer No. 8,220.
2. Withdraw the oil pump shaft No. 8,184. Examine the splined ends of this shaft to see that they do not show excessive wear.
3. Bend down the ears of the nut lock No. 8,536 and remove the nut No. 8,535. This will permit the withdrawal of the upper oil pump screen.
4. Remove four bolts No. 118 which will permit the upper half of the oil pump body No. 8,189 to be removed. This will expose two driven gears No. 8,177 and a driving gear No. 8,187. These gears should be examined as to their fit in the housing. This can best be tested by inserting the drive shaft through its bearing in the housing and laying the gears in the housing in their proper position with their

bearing pins No. 8,179 and No. 8,383 in place. The diametrical clearance of the gears in the housing should not exceed .004 inch. The permissible end play of the gears in the housing is .003 inch.

5. Lift the separating plate No. 8,188. This exposes gears No. 8,178 and No. 8,186 which are part of the oil delivery pump. These also should be examined as to fit in the housing. The above clearances apply to these gears.

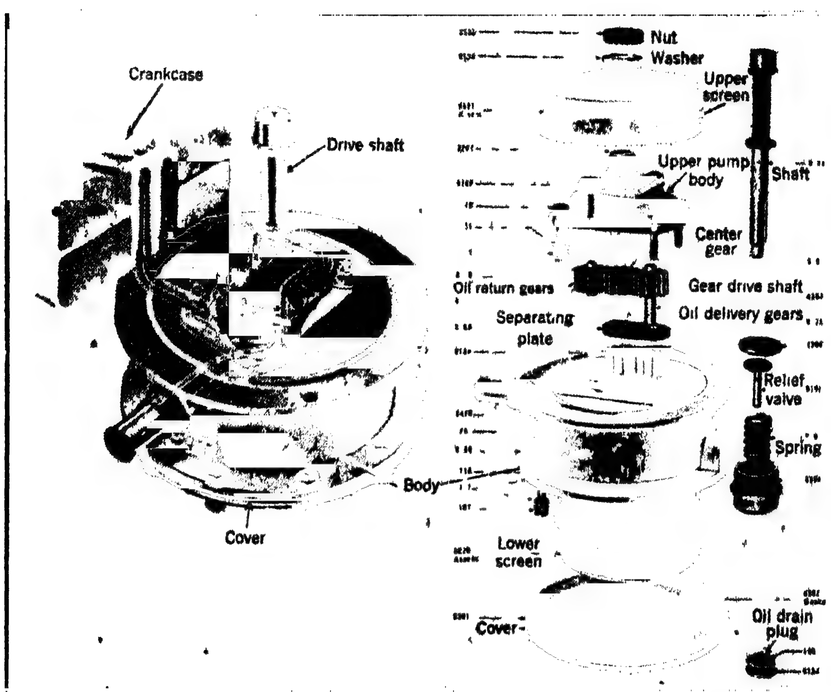


Fig. 451.—Liberty Engine Oil Pump and its Parts.

6. The pressure relief valve may be unscrewed and the valve No. 8,192 and seat No. 8,368 should be examined.

Assembling.—The parts should all be washed carefully and assembled in the following order:

a. The driving gear No. 8,186 and the driven gear No. 8,178 on its pin No. 8,383.
b. The separating plate No. 8,188.
c. The driven gears No. 8,177 and their pins and the driving gear No. 8,187.
d. The oil pump body upper half No. 8,189. **Note—This part of the oil pump housing is located in its proper position by means of two dowel pins No. 174. Care should be taken that these pins are in their proper places.**
The four bolts No. 118 with their washers and nuts may be put in place and tightened up and three of the bolts cottered.

f. The pressure relief valve may now be screwed in against its seat No. 8,368. Lock wire should be passed through the relief valve cage and through the bolt No. 118 which was not cottered.

g. Especial care should be taken that the strainer No. 8,230 is thoroughly cleaned. It may now be put in place and the cover No. 8,381 bolted on. The gasket No. 8,382 between the oil pump body and its cover plate should be replaced if it has been damaged in any way in disassembling.

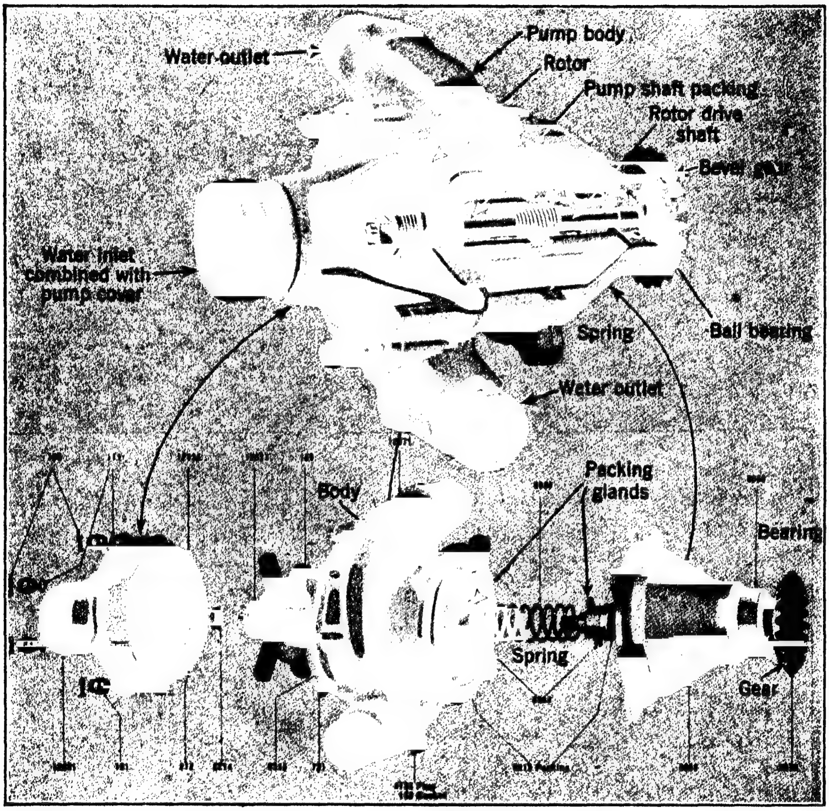


Fig. 452.—Water Circulating Pump Used on Liberty Engine, and Its Parts.

h. Clean the upper screen and put it in place. It will be advisable to install a new nut lock No. 8,536, which may be bent up to lock the nut No. 8,535 after it has been properly screwed down.

Cooling System.—Cooling water is circulated through the Liberty engine by means of a centrifugal pump running at one and one-half times engine speed. The capacity of this pump is 100 gallons per minute at 1,700 r.p.m. The cooling system from the pump inlet to and including the water outlet header will hold 5½ gallons of water. The water pump is provided with a single inlet, the outside diameter of which is two inches, and two

outlets, each one delivering water to a header supplying the right and left hand cylinders respectively. Water is forced into each cylinder jacket tangent to its outside surface. This construction gives the water a whirling motion inside the jacket and insures uniform cooling. The construction of the pump is clearly shown at Fig 452.

The water outlet pipe for each cylinder extends inside the jacket to a point very close to the exhaust valve chamber, which guarantees the proper cooling of the exhaust valve. The cooling water then goes through a passage cored in the intake headers. This serves to warm and further vaporize the incoming gas as well as assist in cooling the water. These passages in the intake headers are connected by two water outlet headers, the final outlet of which has an outside diameter of two inches.

Disassembling and Inspection.—Loosen the hose clamps and pull the hose off the pump water inlet connection.

Loosen the other hose clamps and remove the water inlet manifolds with the cylinder hose connections and the extensions with their hose connections. All rubber hose should be carefully examined for defects—especially on the inside. It sometimes happens that the inside layer of rubber will become loosened and will wholly or partially obstruct the passage.

1. Remove four nuts and washers from flange. This will permit the water pump to be dismounted from the engine.
2. Remove eight nuts No. 101 and the washers No. 111 and take off the water pump cover No. 12,078.
3. Remove nut No. 8,214 from the end of the water pump shaft and draw off impeller by means of the special tool provided.
4. Now lift out the key No. 159 which will permit the water pump shaft to be withdrawn.
5. Examine the water pump shaft. If this is rusted to such an extent that it is very rough it should be replaced. Examine key and key seat
6. Examine the impeller to see that it has not been bent or damaged in in any way.
7. Examine bearing No. 8,056. This should be allowed end play to extent of .005 inch.

To Reassemble.—

1. Put the pump shaft with its bearing in place in the container No. 8,069. It would be advisable to use new packing in the packing boxes. A $\frac{3}{16}$ inch soft round rope packing is recommended for this place, impregnated with a graphite grease. A piece approximately $8\frac{1}{2}$ inches long will be required for each packing box.
2. One piece of this packing should now be wrapped around the shaft and the gland No. 8,059 put in place and pressed down until the packing is properly seated.
3. Compress the coil spring No. 8,058 and fasten it in the compressed state, on opposite sides, by means of twine or soft wire.
4. Now slip the spring in place on the shaft.
5. Put on the second gland No. 8,059 and another piece of the packing No. 8,213.
6. Water pump body No. 12,071 should now be put in place, the key No. 159 inserted in the key seat in the shaft and the impeller No.

12,073 drawn up into place by means of the nut No. 8,214.

7. The wire or string holding the spring in the compressed state may now be cut or drawn out. With the water pump body and the bearing container clamped tightly together the shaft and the impeller should turn freely. The permissible end play in the water pump shaft is .010 inch.
8. Put the gasket No. 8,215 in place. This should be replaced if it has been at all damaged in disassembling.

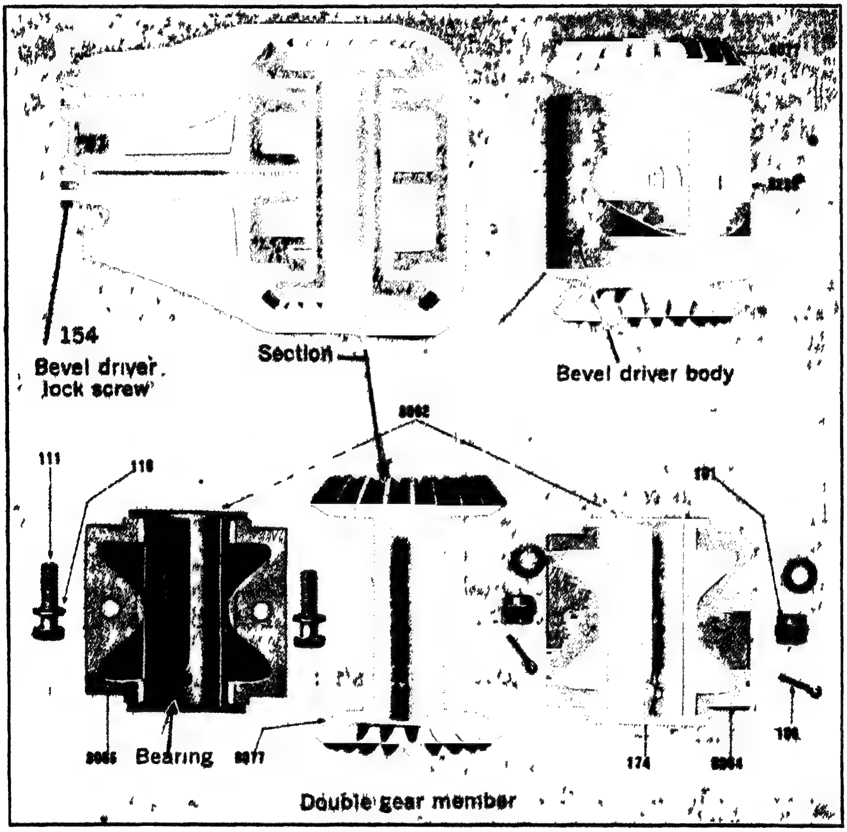


Fig. 453.—The Water Pump Bevelled Driver of Liberty Engines.

9. Complete the assembly by putting the water pump cover in place with its washers and nuts. Be sure that these are all evenly drawn up and properly cotter pinned.

Water Pump Bevel Driver.—Remove screw No. 154 shown at Fig. 453. This will permit the bevel driver and housing assembly to be withdrawn through the opening in the bottom of the crankcase which receives the oil pump. Remove the two clamp bolts No. 116. Examine the shaft of the bevel driver and the bushing in which it runs. Both should be free from scores and indications of cutting. Remove any dirt in the bushing with a scraper. End play of the driver in its bearing should be from .005 inch

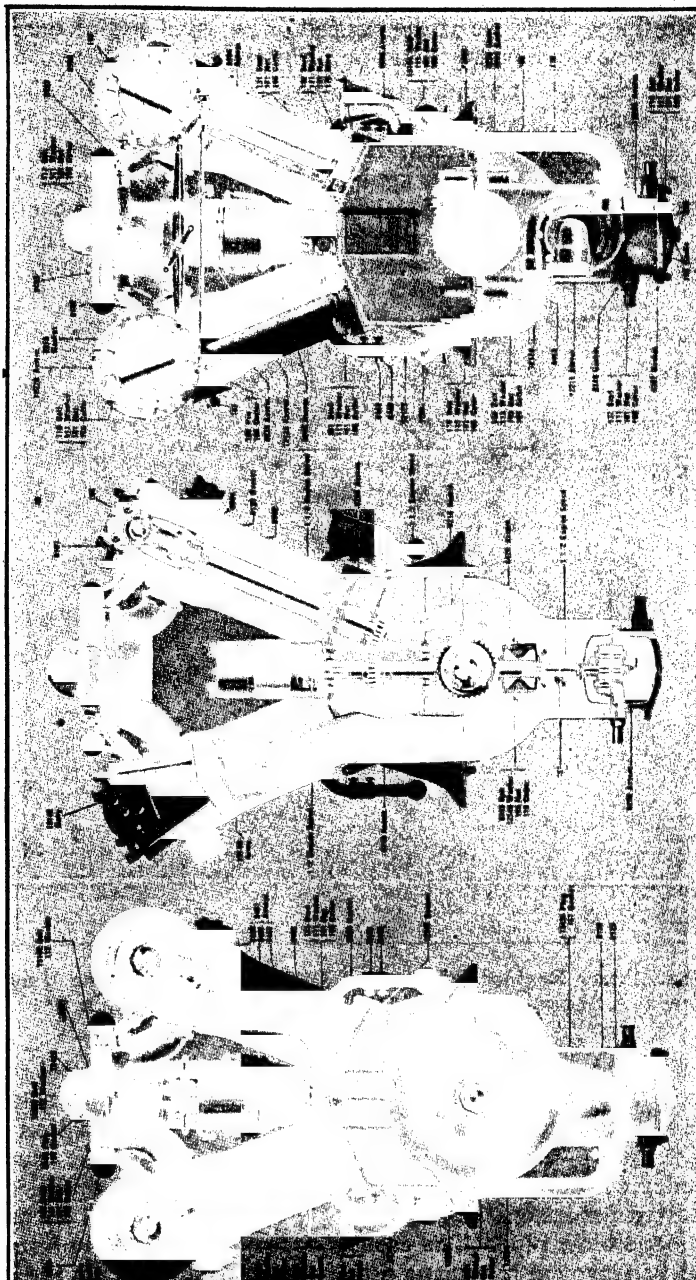


Fig. 454.—End Views of Liberty Aviation Engine Giving Numbers of Parts, which Must be Removed in Dismantling.

to .008 inch. Diametrical clearance should be .0015 inch to .0025 inch. In assembling this unit, the bevel driver should be laid in the housing with the gear with the widest face at the cupped or recessed end of the housing. The two halves of the housing must be properly put together also.

Cylinder Assembly.—Cylinder and valve assemblies may be dismounted by removing the nuts No. 103 and special nuts No. 8,224. (See Fig. 454.) Cylinders should be lifted off carefully and the pistons, when released, should not be allowed to fall over against the sides of the crankcase. After removing the cylinders the nuts should be put back on the studs and screwed on just far enough to prevent them from being lost. This will not only protect the studs but will prevent confusion in reassembling. If some

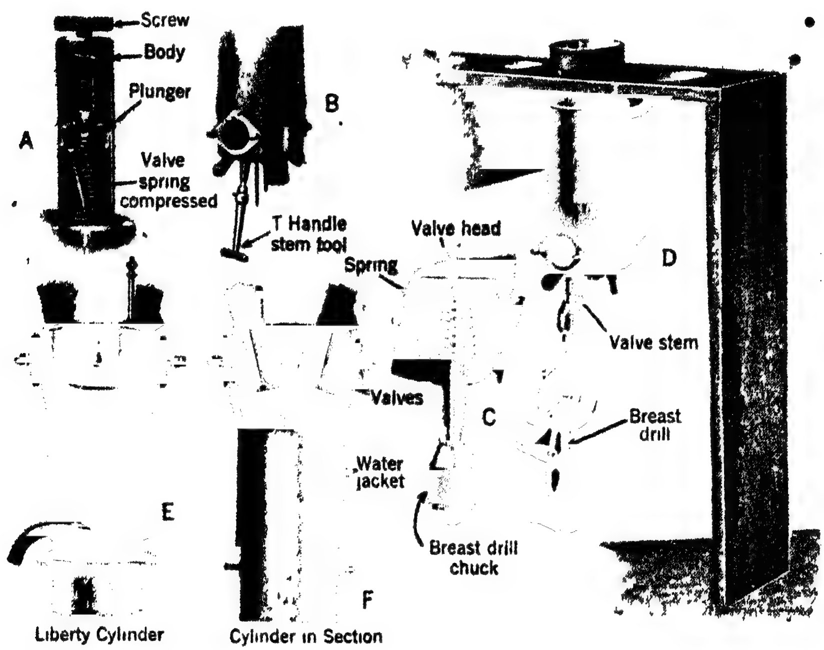


Fig. 455.—The Liberty Engine Cylinder and Method of Grinding the Valves. The View at A Shows Special Valve Spring Compressing Tool in Use.

old pieces of hose are available, short sections should be slipped over two studs on opposite sides of each cylinder opening. The cylinder serial numbers are stamped on the edges of the base flanges. The location of each cylinder on the engine (as "1 Left" etc.) is also stamped on the base flange. If water under pressure of ten pounds is available, each cylinder jacket should be tested for leaks. Any small leaks may be repaired by means of an oxy-acetylene blow pipe. The carbon deposit on the inside of each cylinder should be removed by means of soft scrapers. Examine the cylinder barrels carefully for scores or scratches.

Remove the Valves.—To do this the valves must be held up against their seats and the springs compressed sufficiently, by means of a valve

spring compressor supplied in the tool kit, as shown at Fig. 455 A, to permit the lower collar being moved down far enough to uncover the valve spring collar keys. These keys (two parts) may now be lifted out and the springs and valves removed. The same inside spring is used on both intake and exhaust valves. The outside spring may be identified by the fact that the exhaust valve spring has ten coils while the inlet valve spring has twelve coils. Also the exhaust valve spring exerts a pressure of 23½ pounds when compressed to a length of 2¼ inches. The valves for each cylinder have the cylinder serial number etched on them and are marked "Ex" (exhaust) and "In" (inlet) respectively.

Clean the valves carefully and if the valve stems are gummed up with burned oil they should be dressed down with fine sandpaper. Examine the valve seats in the cylinder and the face of the valves which comes in contact with these seats. The valves should be tested for gas tightness. This can best be done by inverting the cylinder with the valves in place and pouring a small quantity of gasoline in the cylinder. Watch for seepage around the valve. If the valves show any leak, they should be carefully ground in. The cylinder, for this operation, should be held in position by means of the flange at the bottom. An easily made arrangement for holding the cylinder for valve grinding is shown at Fig. 455 D.

Note that in the illustration Fig. 455 C a light spring is inserted under the valve to partially counterbalance the weight of the tool by means of which the valve is turned. An efficient valve grinding tool, as shown at Fig. 455 B, will be found in the tool kit. In grinding the valve it should not be revolved, but should be rocked backward and forward and frequently lifted off its seat and its position on the seat changed in order to distribute the abrasive evenly and to prevent "scoring" the valve. Valves should not be ground any oftener than is absolutely necessary and then only enough to "clean up" the seat. If a valve is pitted or warped to such an extent that it is necessary to grind it heavily, care should be taken that any ridge or shoulder formed on the edge of the valve seat be dressed down with a fine mill file. The abrasive should be carefully washed off the valve, the seat and the inside of the cylinder. Test seating of valve with Prussian blue.

Any valves which are badly warped should be replaced and the valve seat in the cylinder should be trued up with a valve seat reamer before grinding in the new valve. Replace the valves, the lower collars, the springs, the upper collars and the collar keys.

Dismounting Pistons.—Remove the piston pin retainers by either one of the methods shown in Figs. 456 D or E. Press out the piston pins with a brass drift supplied in the tool kit. The fit of the piston pin in the piston should be from a free running fit without play to a light drive fit. It is preferable to have the piston pin a free fit in the piston than to have it drive out too hard. Clean the carbon deposit off from the piston head both on the top and on the under side. The polished surface of the top of the piston head should not be marred or scratched if it can be avoided.

Bearing Surface.—Examine the piston for scores. It is very likely that the pistons will show scratches which were caused during the first run in of the engine. It is difficult to draw a line of distinction between what is

termed a scratch and a score. A piston should not be discarded unless the scores extend past the piston rings and seem to be of recent origin and of appreciable depth. Examine the piston for even bearing on its outside surface. If any piston shows excessive wear on one side at the bottom and not at the top, it is an indication that the connecting rod is twisted or bent. This rod should be tagged and straightened up before assembling.

Rings.—Examine the piston rings for even bearing on the outside surfaces. The ring should be a free fit in the grooves, and should not be so loose that any shake is noticeable against the sides of the piston lands. Inspect condition of ring grooves through the gap as to carbon deposit. If the carbon is soft and not of great amount it may be wiped out with a soft

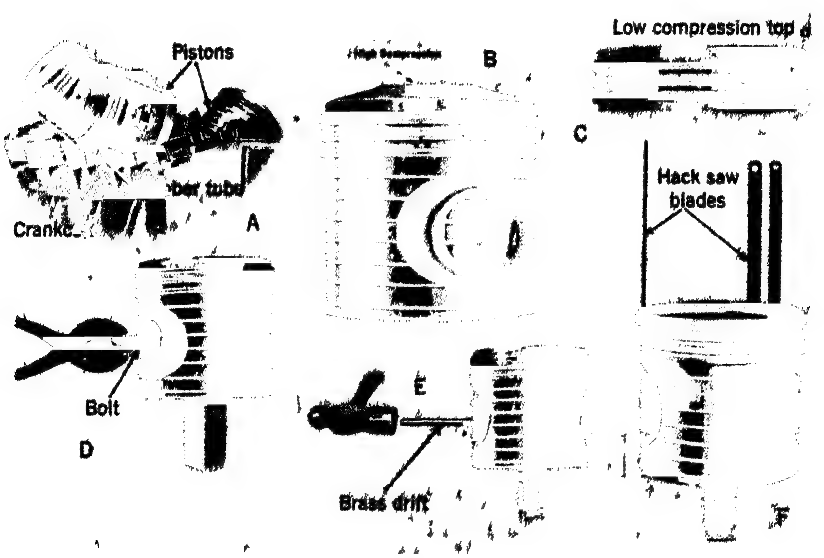


Fig. 456.—Illustration Showing Method of Handling Liberty Engine Pistons. A—Use of Rubber Tubing on Studs to Prevent Marring Piston. B—High-Compression Piston has Domed Top. C—Low-Compression Piston has Practically Flat Top. D—Method of Removing Ring Pin Lock by Bolts. E—Method of Removing Ring Pin with Drift and Hammer. F—How to Remove Piston Rings without Breaking Them.

rag over a splinter of wood inserted through the gap in the ring. If the amount of carbon is excessive and caked hard, the ring should be taken off.

Extreme care should be exercised in removing the rings which may be done as shown at Fig. 456 F. They should not be expanded more than is absolutely necessary to pass them over the top of the piston. As the piston material is comparatively soft, care should be taken not to mar or scratch its outside surface. Ring grooves should be wiped out with a soft cloth moistened with gasoline, and any carbon caked in these grooves may be scraped out with a piece of wood. It is preferable to put back the old rings if the wear has not been too excessive, than to fit new rings which have not been run in.

In refitting the rings in the grooves the same care should be exercised as in removing them in order that the piston surface may not be scratched

or marred. The gap between the ends of the ring should be not less than .025 inch when the ring is fitted in the cylinder. The pistons are marked as to their location in the engine 1L, 1R, etc. These marks are stamped in the depression on the side of the piston. In reassembling pistons on rods, the marked side should be towards the distributor end of the engine. The number of ounces which each piston weighs over three pounds is also stamped in this depression.

If it was found necessary to replace any pistons, new ones of exactly the same weight should be selected from stock. **Caution—Great care should be exercised in handling pistons to prevent them from being subjected to any sort of rough usage which would be likely to spring them "Out of round." Do not allow piston with rings to slip by the counter bore at the top of the cylinder. It is almost impossible to compress the rings enough to push the piston down again without damaging the surface of the piston or the cylinder wall. To apply or remove piston rings, use hack saw blades with the teeth ground off and all the sharp edges rounded so they will not cut.**

The bearing surface on the piston illustrated at F, Fig. 456 indicates a bent or twisted connecting rod. Note that the heaviest bearing is at the top right and bottom left. The lighter scores are characteristic of aluminum pistons. The piston shown should not be condemned on account of these scores.

Connecting Rods.—1. After the pistons are all removed from the connecting rods then:

2. Turn the crankcase bottom side up and take out all the small bolts which fasten the two halves of the crankcase together.

3. Lift off the bottom half of the crankcase being careful to raise it steadily and evenly so as not to spring or bend the crankshaft bearing bolts.

4. Examine the connecting rod bearings for side play. The bushing carried by the forked rod should have from .010-inch to .020-inch side play on the crank pin. This may be checked with a thickness gauge. The plain end connecting rod should have from .004-inch to .008-inch side play in the forked rod.

5. Remove each pair of connecting rods by first taking off the nuts No. 13,251 and the cap of the plain end rod, then take off nuts No. 13,220 and the caps of the forked end rod as shown at Fig. 457.

Note:—That the caps and rods are each numbered at the joint on one side. The caps should always be replaced on the rod with like numbers adjacent. The bushings are also marked with small numbers on the flange. These numbers should always be adjacent and on the side of the rod which bears the cap numbers. Each connecting rod is also numbered on the web near the piston pin end. The upper number at this point indicates the serial number of the crankshaft to which the rods have been fitted. The lower number indicates the position of the rod in the engine, as 1, 2, 3, etc., numbering from the distributor end of the engine.

6. Examine the crank pin bushing carefully. It should show a 75 per cent bearing free from dirt and scratches or scores. Any dirt or foreign matter should be carefully scraped out. The outside of the crankpin bush-

ing on which the plain end rod bears should also be carefully examined. Any roughness on its surface should be dressed down with a fine mill file. Oil grooves and oil holes should be cleaned out.

Figure 458 A shows a perfect surface with a 100 per cent bearing. Figure 458 B is also a 100 per cent bearing. The black spots are aluminum chips imbedded in the babbitt. The pits or depressions in the surface were caused by aluminum chips or dirt which have been wiped off or fallen out. A bearing in this condition should be very lightly touched up with a scraper.

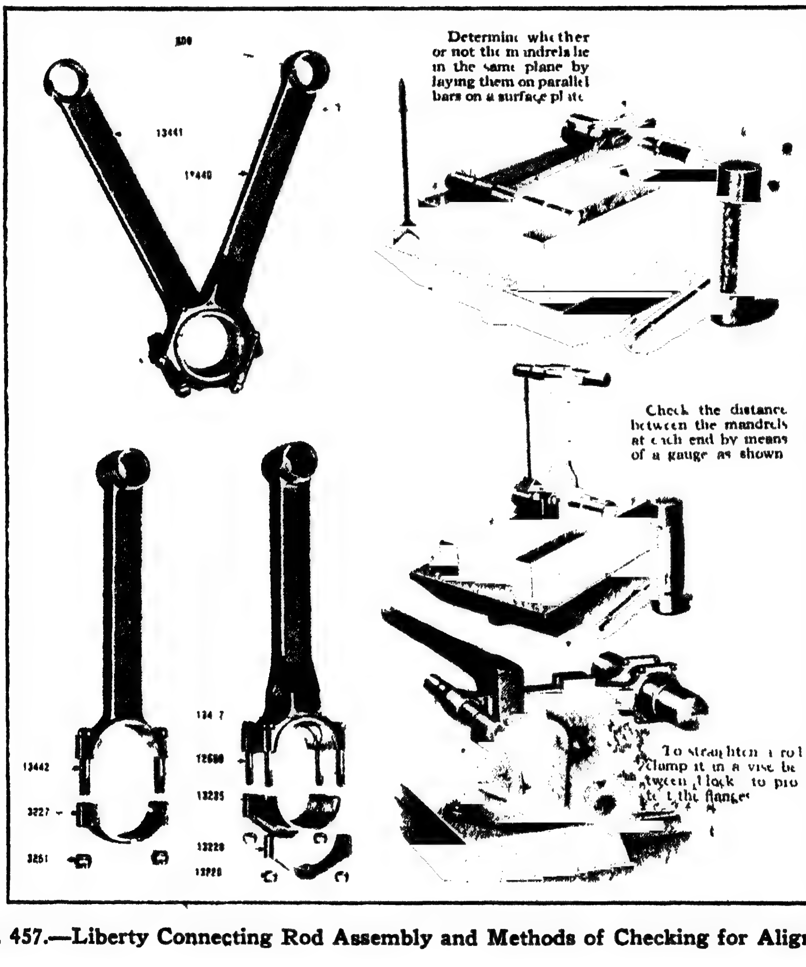


Fig. 457.—Liberty Connecting Rod Assembly and Methods of Checking for Alignment.

Figure 458 C shows a good bearing surface The scratches in the vicinity of the oil hole were caused by sand or grit carried in by the oil. These scratches should be smoothed up with a scraper. Figure 458 D is an example of cracked babbitt. This bearing was evidently fitted too tightly as indicated by the burnished surface just above the center. Excessive pressure at this point has caused the babbitt to "drag" or "wash" in the direction of rotation of the crankshaft. This bearing would probably be

good for another twenty hours' service in spite of the displaced metal.

Fitting Connecting Rod Bearings.—If the bearing has been damaged or shows wear to such an extent that it is advisable to replace it, the new bushings should first be fitted in the forked end rod. Be sure that the bushing seats properly in the rod and that the dowel does not hold it away at any point. The caps of the forked end rod should be put in place and drawn up tightly. Examine the joints between the cap and the rod and between the two halves of the bushings. Caps and bushings should bear equally hard at the joints. The bushing should then be scraped to a free fit on a mandrel .03 inch to .004 inch larger than the crankpin. The cap should now be removed and the rod tried on the crankpin on which it is to run. The ends of the bushing should be dressed off with a fine mill file and a sufficient amount removed to permit from .010-inch to .020-inch side play. Touch up the radius at each end of the bushing with a scraper until it clears the fillet of the crankpin. Test this point by coating the crank pin and each fillet lightly with red lead or Prussian blue.

Clamp the rod with its bushing on the crankpin. Revolve it on the crankpin two or three times, meanwhile pressing it first toward one end of the crankpin and then toward the other end. Remove the rod and carefully scrape off any high spots to which the color has been transferred. It is very essential that these bushings do not bear on the fillet.

7. With the bushing clamped in the forked end rod dress down the outside surface at the joints with a fine mill file until they are perfectly smooth. Coat the bearing surfaces of the plain end rod lightly with red lead or Prussian blue and fit it in place on the bushing. Any high spots on the bearing to which the color is transferred should be dressed down until the plain end rod has .005-inch to .0065-inch diametrical clearance. The roundness of the bushing should meanwhile be tested with micrometers.

To determine the diametrical clearance which a connecting-rod bearing has, the crankpin and the outsde of the connecting-rod bushing should be carefully measured with micrometers. The connecting rods should then be clamped on test mandrels, the diameter of which may readily be determined. Connecting-rod bearings should be fitted only at repair shops equipped with a complete set of such mandrels.

8. Examine the bushing at the top end of the connecting rod. This should show a burnished bearing surface free from scratches or scores. The radial clearance of this bushing and its proper piston pin should be .001 inch. If it has been necessary to replace this bushing, the new one should be oiled on its outside surface and carefully pressed into place under an arbor press or in a vise. After being pressed into place it should be reamed with an expansion reamer until it has the proper clearance on the piston pin.

Aligning Connection Rods.—When new bushings are fitted in a connecting rod, or if an inspection of the piston would indicate that the connecting rod was twisted or bent, the rod should be checked for alignment in the manner shown at Fig. 457. A snug fitting mandrel should be put through the bearing in each end of the rod. These mandrels should be at least twelve inches long. The rod with the mandrels in place should be laid on a surface plate and the mandrels checked as to whether or not they

are parallel and in the same plane. A twisted or bent rod may be straightened by clamping one end of it in a heavy vise and springing it with a special forked-end bending bar. A rod which has been badly bent should not be used, nor should it be heated in order to facilitate straightening.

Crankshaft.—After the connecting rods have been inspected, overhauled and laid to one side, the crankshaft may be lifted out of its bearings. It will appear as shown at Fig. 459. All bearing surfaces on the

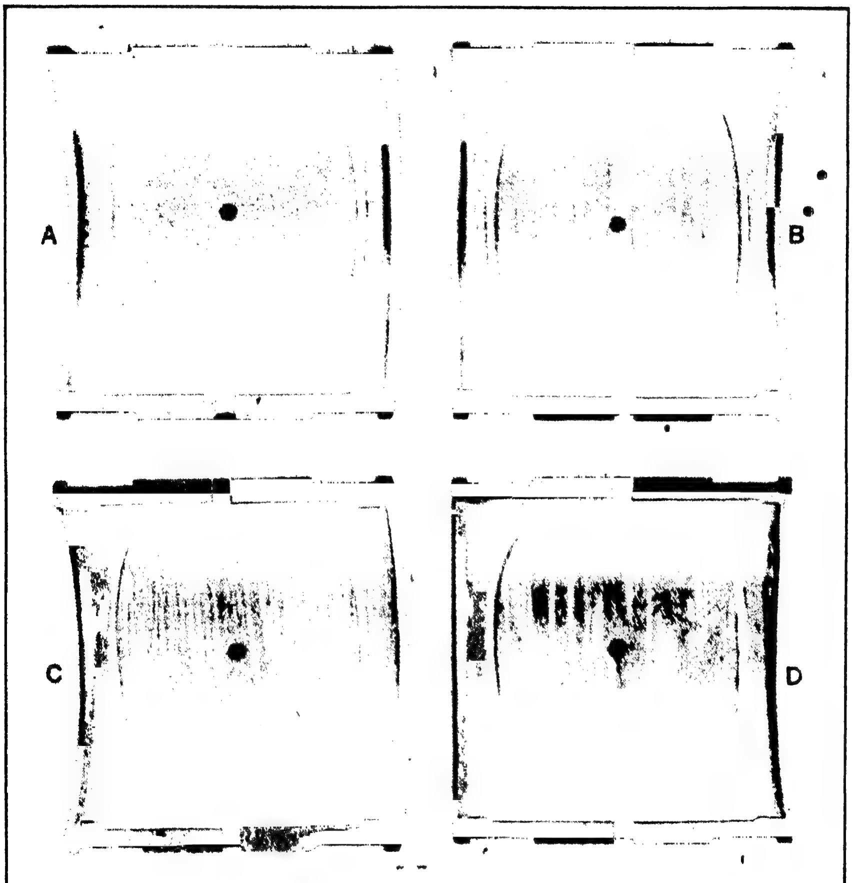


Fig. 458.—Various Stages of Wear in Crankpin Bushings of Liberty Engines Outlined. A and C Show Good Bearing Surfaces. Surface at B is Embedded with Small Metal Particles. At D, the White Metal Lining Has Picked Up.

crankshaft should be carefully examined. Unless one or more of the bearings have been completely destroyed, the crankshaft surface will be in good condition. If any crankpin or bearing surface on the crankshaft is roughened or scored it should be dressed down with a fine mill file and fine sandpaper and oil, meanwhile being tested as to roundness with a micrometer. This work can only be done by an experienced mechanic.

Examine the ball thrust bearing. This should show not to exceed .008-inch end play. Unless the thrust bearing shows damage or excessive

end play after being washed, it should not be necessary to remove it. Be sure that it is packed with grease before engine is finally assembled. Examine the lock screw through the thrust bearing retaining nut. Be sure that it is tightly drawn up and that the lock wire sets fairly in the slot. If it is considered advisable to remove the propeller hub, follow the special instructions given. Examine the tapered end of the shaft on which the propeller hub fits. If this has been marred or roughened up in any way the key should be removed and the rough spot smoothed up. The hub should then be lapped on the end of the shaft. Examine the condition of the key and its fit in the shaft. Examine the crankshaft gear to see that it does not show excessive wear.

Examine the plug in the propeller end of the shaft and the plugs in the ends of each main bearing and crankpin. They should all be tight and show no indications of oil leakage. Clean out the oil cavities and passages in the crankshaft and crankpins. This can best be done by filling each one with gasoline or kerosene, allowing it to stand for a short time and then blowing out with compressed air. Unless the crankshaft, thrust bearing or propeller hub were found to be defective, no attempt should be made to remove the hub.

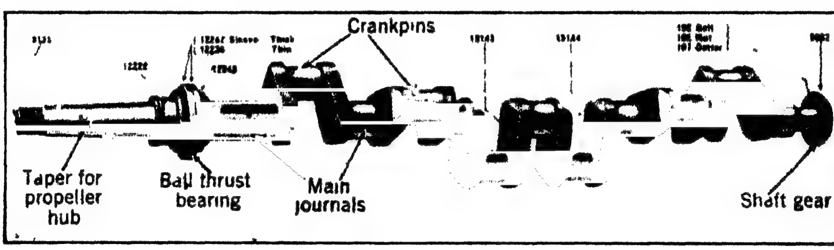


Fig. 459.—Crankshaft of Liberty Engine.

Removing Propeller Hub.—If it should be necessary to take the hub off, proceed as follows: Remove the lock wire and unscrew the lock nut (the largest one) and then the retaining nut entirely out of the hub. (See Fig. 460.) Oil the threads of both liberally.

Now screw in the retaining nut (the smallest one) as far as it will go and then back it out five turns. Screw in the lock nut until it touches the shoulder on the retaining nut. Continue to screw in the lock nut, meanwhile holding the retaining nut and the crankshaft stationary. This will draw the hub off the shaft. Or screw out the retaining nut, meanwhile holding the lock nut and the crankshaft stationary.

It may be necessary to heat the hub in order to start it. If a blow torch is used for this purpose, heat should be applied to the barrel of the hub in the vicinity of the flange and as evenly as possible.

Fitting Propeller Hub.—Propeller hubs should be "lapped" to a fit on the shaft. The work should be done in such a manner that the hub will be about .001 inch tighter at the large end of the taper than at the small end. This result can be accomplished as follows:

1. Lap the hub to a fit on the full length of the taper.

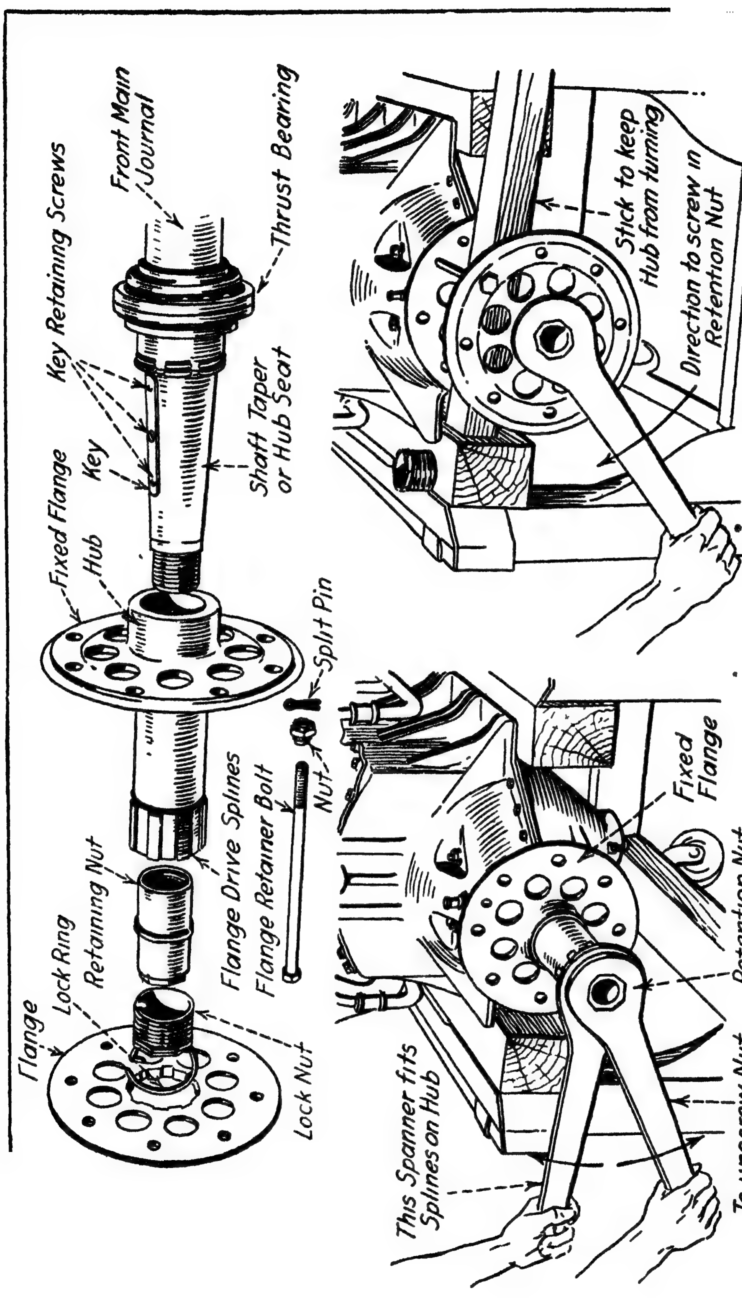


Fig. 450.—How the Propeller Support Hub is Mounted on the Crankshaft of Liberty and Similar Engines.

2. Wash off the abrasive thoroughly from both hub and shaft with gasoline.
3. Lubricate the large end of the taper with a light machine oil.
4. Apply the abrasive to the small end only of the taper and lap the hub on again.
5. Check the progress of the work occasionally by washing off the oil and abrasive and testing the fit with Prussian blue.

The final result should be such that the color will be spread thin at the large end but should remain heavy at the small end of the taper.

Mounting Propeller Hub.—Immerse the hub in boiling water or oil for two or three minutes, then tap lightly into place on the shaft and immediately apply the retaining nut and draw it up tight. Now apply the lock nut and draw it up only tightly enough to insure a snug bearing against the flange of the retaining nut. Apply the lock wire, making sure that the tongue is long enough to reach through both nuts. **Caution—Be sure that there is a minimum clearance of .010 inch between the top of the key and the bottom of the keyway in the hub.**

Crankcase.—Examine all bearing bushings in both the upper and lower halves of the crankcase. The bushings should show a 75 per cent bearing and should be free from dirt, scratches and scores. Should it be necessary to replace any of these bearings, new bushings should be carefully fitted in each half of the crankcase. They should then be carefully scraped to a free fit on the mandrel .0025 inch to .00325 inch larger in diameter than the crankshaft. In scraping the bushings to a fit on this mandrel, care should be taken that the case is supported and clamped down in such a manner that it will not be sprung or twisted out of shape. It is advisable to clamp the case at only one point and that only rigidly enough to hold it steady. Contact between the two halves of the bushing and the two halves of the crankcase should be equal when they are clamped together.

Examine the crankshaft bearing bolts and see that the nuts at the lower end of these bolts are drawn up tightly and properly cotter pinned. Examine the threads on the top of these bolts to see that they are not stripped. Wash the inside of the case thoroughly and clean and blow out all oil passages with compressed air. Extreme care should be exercised in carrying out this part of the work. **Caution—The joint between the two halves of the case is "lapped" and the parts should be carefully handled to avoid possibility of marring the surfaces.**

To Assemble the Engine.—Crankcase Upper Half—1. Place the upper half of the crankcase in the inspecting stand bottom side up. 2. Wipe out all bearings carefully. 3. Oil all bushings with fresh clean oil and lay the crankshaft with its thrust bearing in place. Be sure that the thrust bearing is properly seated in the case and that the sleeves do not permit end play. **Note—The thrust bearing sleeves are made in different thicknesses, No. 12,252 being the thick one and No. 12,238 the thin one. A combination of these sleeves may be obtained, which will insure the proper fit in the case.**

4. **Connecting Rods.**—The connecting rods may now be assembled on the crankshaft. The rods are numbered on the web at the piston end 1, 2, 3, etc., indicating their location on the crankshaft beginning at the gear end.

The side of the rods on which the numbers are stamped should be toward the gear end of the shaft. All forked rods should be so fitted that they will be on the right hand side of the engine looking from the gear end toward the propeller end when the engine is right side up. Crankpins should be liberally oiled before the rods are put in place, as should also the outside of the connecting rod bushings before the plain end rods are assembled. The hollow crankpins and main bearings should be filled with clean oil. Draw up all connecting rod bolts snugly and cotter pin them. **Note—The cotter pins in the bolts through the forked end rod should be inserted so that the heads will be on the inside or toward the plain end rod. Otherwise the bent-over ends of the cotter pins will interfere with the plain end rod. Be sure that none of these bolts are overlooked.**

Crankcase Lower Half.—The lower half of the crankcase may now be put in place. 5. Put the washers and nuts on the front and rear bearing bolts and draw them tight.

6. **Water Pump Bevel Driver.**—Put the water pump bevel driver with its bushing in place and fasten with the lock screw. Test the back lash of the water pump bevel driver gear and the crankshaft gear. This should be .005 inch minimum and .010 inch maximum. If these gears mesh too tightly, shims No. 8,502—.002 inch thick may be placed between the crankshaft, gear and the crankshaft flange in sufficient number to produce the proper back lash.

7. Now turn the whole assembly right side up and complete the installation of the bearing bolt washers and nuts. Draw up these nuts snugly and cotter pin them.

8. Install bolts No. 116 and tighten and cotter pin them.

9. **Pistons.**—Proceed to fit the pistons on their proper rods. Remember that the pistons are marked right and left and numbered 1, 2, 3, 4, 5 and 6 and that the pistons should be so placed on the rods that the numbers are toward the gear end of the engine. The piston pins should be oiled before being pressed into place.

10. Insert the piston pin retainers No. 12,547. These should be a light tapping fit in the piston bosses.

11. **Cylinders.**—Oil the inside walls of the cylinders and the outside of the pistons. The cylinders are numbered 1, 2, 3, 4, etc., and are marked right and left, indicating their position on the crankcase. These marks will be found on the edge of the base flange near the water inlet. Be sure that the gasket No. 12,346 is in place on the crankcase studs. The crank should be turned over so that the piston on which the cylinder is being placed is at the top of its stroke. The rings should be compressed in the grooves as the cylinder is slipped down over them.

12. Push the cylinder carefully down into place, put on the nuts No. 103 and run them down on the studs, but do not tighten them.

13. After all the cylinders are in place, put on the gaskets No. 8,175 and the intake headers No. 12,045. **Caution—Insert loose-fitting wooden plugs in sparkplug holes, to prevent any small parts from falling into cylinders.**

14. **Intake Headers.**—Now put on the intake header stud washers No. 113 and the intake header nuts No. 103. The intake header nuts and the

cylinder to crankcase nuts on each set of three cylinders should all be tightened up at the same time, i.e., each nut should be drawn down one after another about one-fourth turn at a time until all are tight with an equal tension. (See Fig. 454.)

15. The special nuts No. 8,234 may now be put in place and tightened up with a special long shanked socket wrench.

16. Tighten all nuts and properly cotter pin them.

17. Put the gaskets No. 8,172 between the intake header and the water inlet elbows No. 8,173 and tighten these up with the cap screws No. 201. The gaskets No. 8,172 should be soaked in water before putting them in place. Leave three screws No. 201 on each side (one at each end of the engine and one about midway between these two) loose until the cable tube has been installed.

18. **Outlet Headers.**—Put the water outlet header gaskets No. 8,153 in place and bolt on the front and rear water outlets No. 12,402 and No. 12,401 respectively. Before these water outlets are bolted down the extension tube No. 12,363 and the two pieces of hose No. 12,128 should be slipped in place.

19. Now fasten down the water outlets with nuts No. 101 and cotter pin them.

20. Slip the carburetor to intake header bolts No. 8,398—No. 8,399 in place.

21. **Carburetors.**—Attach the two carburetors. Be sure that the unions and gaskets are in place. With respect to these gaskets it is advisable to fit new ones after an engine has been overhauled. Care should be taken that the carburetors are mounted in their proper position on the engine (propeller end and gear end) and that the air intake scoops open toward the propeller end.

22. **Throttles.**—The carburetor throttle coupling shaft No. 12,498, shown at Fig. 463 A, may now be put in place and the two sets of throttle synchronized as follows: Screw out the throttle stop screw on each carburetor and, holding them firmly in a closed position, screw up the adjusting screws No. 13,416 until they just touch each side of the lever No. 12,498 and lock the screws by means of a wire through the head of each. Connect up the altitude adjustment coupling rod in such a manner that both altitude adjustments will get the full throw in each direction.

Timing Engine.—A part of the equipment of every Liberty engine is a timing disc, so designed that it may be mounted on the propeller hub as a permanent fixture or used only in the hangar or repair shop as a means of checking the setting of valves and distributors. To use this timing disc which is shown at Fig. 461 A, proceed as follows:

1. If the timing disc is not already mounted on the propeller hub, install it in such a manner that the dowel in the propeller hub flange enters the dowel hole in the disc. It may be clamped in this position by means of two bolts through the propeller hub bolt holes.

2. Remove the sparkplug from the propeller side of No. 6L cylinder.

3. Insert a pencil or scale through the sparkplug hole and turn the engine over until the piston on its up stroke touches the pencil and causes it to ride up. Continue to turn the engine over slowly until the piston as in-

dicated by the travel of the pencil stops moving upward and is just about to start down. This will be approximately the top dead center.

4. Allow the crankshaft to remain in this position temporarily and clamp the timing pointers, which will be found in the tool kit, under the special cylinder base flange nuts so that the pointers extend over the edge of the timing disc as shown in the illustration.

5. With the end of the pencil resting on the top of the piston make a mark with a knife blade about one-half inch above the edge of the sparkplug hole.

6. Turn the engine over in a forward direction until the pencil has moved down so that the mark is even with the top edge of the hole, and with a piece of chalk or a pencil mark the disc in line with one of the pointers.

7. Turn the engine backward until the pencil has moved up and down to the point where the mark is again even with the top of the sparkplug hole, and mark the disc in line with the pointer.

8. With a pair of dividers find the point midway between the two marks on the disc. This point will indicate the exact dead center of No. 1 and No. 6 cranks and should be marked with chalk or pencil.

9. Turn the engine over until this dead center mark is in line with the pointer. Allow the crankshaft to remain in this position and

10. Reset the pointers so that they come in line with the dead center marks stamped on the disc.

11. Turn the engine over in the direction of rotation through ten degrees as indicated by the scale on the disc.

Neutral Point.—The crankshaft is now set on the neutral point of No. 6 left cylinder and the firing point—spark retarded of No. 1 left cylinder.

Reference to the timing diagram at Fig. 461 B will explain what is meant by "neutral point." It is the point ten degrees past the top dead center which marks the beginning and end of the cycle of operations. The exhaust valve closes and the inlet valve opens at this point.

Mount the generator drive shaft assembly No. 8,210, being careful that the gasket No. 8,347 is in place and a sufficient number of shims No. 8,511 (.002 inch thick) to insure proper mesh of the generator drive shaft lower gear with the crankshaft gear. These gears should have a minimum back lash of .005 inch and a maximum of .010 inch.

Mount the two camshaft drive shafts No. 8,208, meshing the gears in such a manner that the mark on the splined coupling is "fore and aft" or parallel with the center line of the engine.

Now mount the camshaft housing assemblies.

If it was not found necessary to replace either the camshaft or gear, be sure that the marked teeth on both gear and pinion are in line. This should bring the mark on the splined end of the drive shaft "fore and aft." The assemblies may now be set in place with the splined coupling marks in line.

Note—All marks for both right and left cylinders are located with No. 1—6 cranks ten degrees past left dead center.

Complete the installation of these assemblies by replacing the washers No. 112 and the nuts No. 102 and properly cotter pinning them.

Slip the felt washers No. 8,068 into place and tighten up the stuffing boxes No. 8,066.

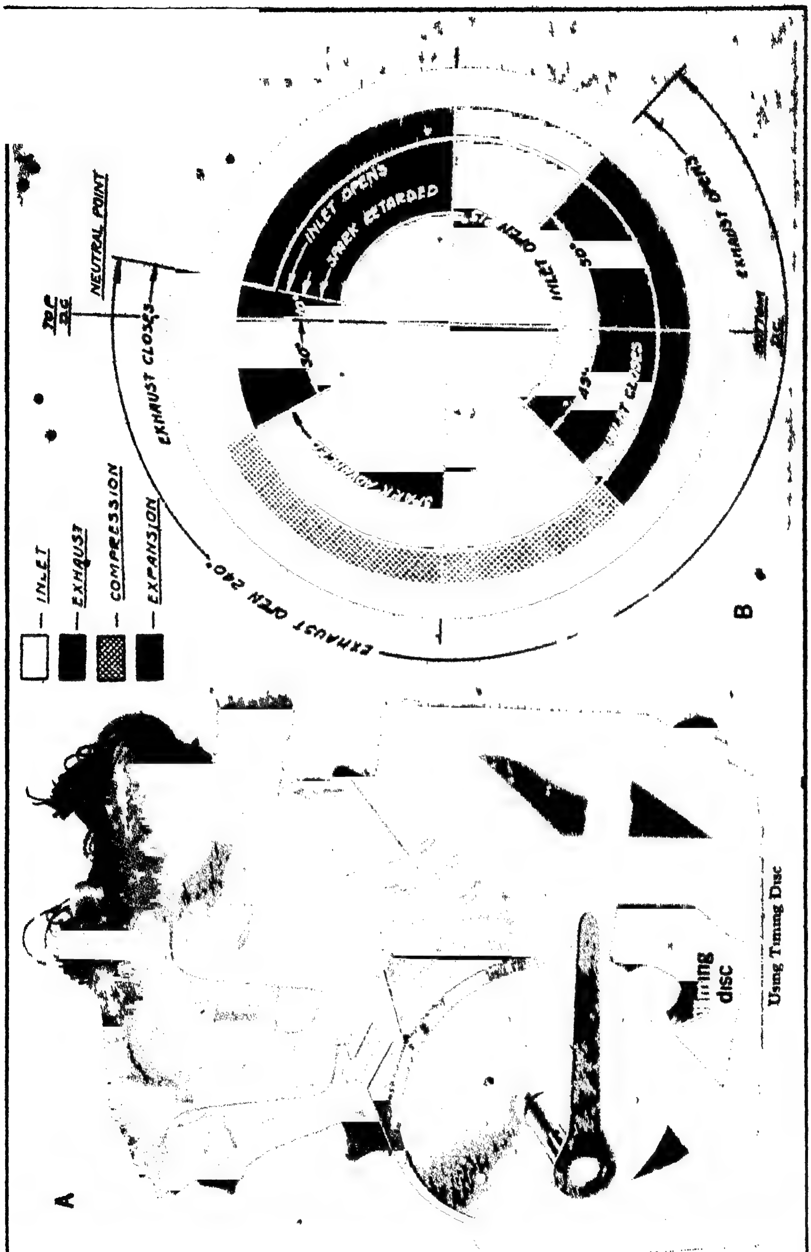


Fig. 461.—Diagrams Illustrating Timing the Liberty Engine. A—Use of Double Trammel and Timing Disc. B—Valve Timing Diagram.

Tappet Gap and Firing Point.—Test the gap between all tappets and the valves which they operate The tappet gap for each cylinder should be checked when that cylinder is on the firing point. The firing point of No. 1 cylinder is the neutral point of No 6 on the same side. The firing point of No 2 is the neutral point of No 5 The firing point of No. 3 is the neutral point of No. 4 The firing point of No 4 is the neutral point of

No. 3 The firing point of No. 5 is the neutral point of No. 2. The firing point of No. 6 is the neutral point of No. 1. It will be noticed that the sum of the numbers of these pairs of cylinders is always seven. For example—to find the firing point of No. 4 cylinder, turn the engine over, meanwhile watching the No 3 exhaust valve. When this valve has just closed and before No. 3 inlet valve has opened, the neutral point of No. 3 cylinder will have been reached. This will be the firing point of No. 4.

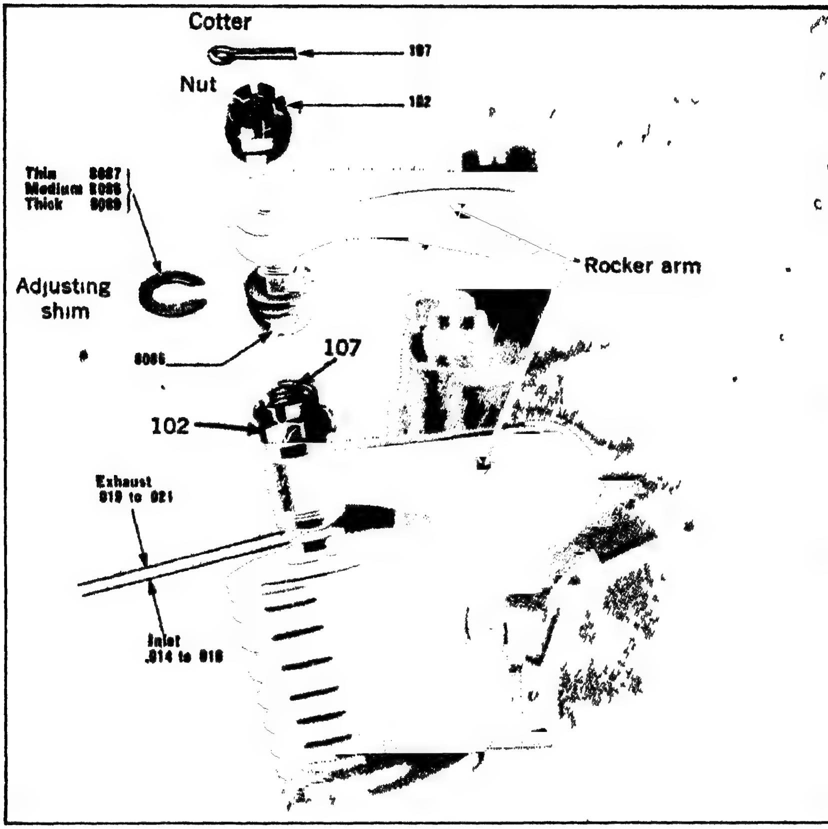


Fig. 462.—Liberty Engine Rocker Arm and Method of Adjusting Clearance Between Valve Stem and Tappet.

With the engine cold the clearance between the inlet valve tappets and the valve stems should be .014 to 016. The clearance between the exhaust valve tappets and valve stems should be 019 to 021. This clearance should be adjusted by adding or taking out shims under the tappet head as shown at Fig. 462. These shims are made in varying thicknesses The thick shim No 8,089 being .015 inch thick, the medium shim No. 8,086 being .008 inch, and the thin shim No. 8,087 being .003 inch. The combination of these shims will permit of a very accurate adjustment of the gap. Be sure that the shims are properly placed and that the nuts on the tappets are tightly drawn up and cottered.

If it was found necessary to replace either the camshaft or the camshaft gear, the gear should have been left unmounted. In this case proceed as follows: 1. With No. 1—6 crank set ten degrees past the left dead center, the marked splines on the camshaft drive shaft set "fore and aft," the marked splines on the upper camshaft drive shaft gear should be toward the observer and on the center line of the cylinders.

2. Without moving any of this assembly rotate the left camshaft in a clockwise direction until the No. 6 exhaust valve is just closed and the inlet valve is just about to open.

3. Now mesh the camshaft gear in such a manner that the teeth and the flange bolt-holes will line up perfectly. The camshaft gear has 48 teeth and is bolted to the flange by means of seven bolts. This will permit an adjustment of one-seventy of one tooth space or two and one-seventh degrees of crankshaft travel.

4. Tighten up two of the camshaft gear bolts and check the tappet clearance on all left cylinders.

5. **To Advance or Retard Valve Setting.**—Now check the opening and closing of the exhaust and inlet valves as shown on the timing diagram. If it is found that the valves are late in opening and closing, the number of degrees should be noted and the camshaft gear moved one or more holes in the direction of rotation without moving the camshaft drive shaft or the camshaft. Remember that for each hole moved forward, the camshaft is advanced two and one-seventh degrees of crankshaft rotation. If the valves are found to open early, set the camshaft gear backward one or more holes.

Always check valve timing by turning engine in forward direction of rotation so as to take up all back lash in gears and lost motion in couplings.

After the gear has been properly located, set the left distributor driving flange over the bolts in such a position that the marked notch is in line with the marked tooth on the drive pinion.

Now tighten up and cotter-pin the bolts and mark the gear in line with marked tooth on the drive pinion. To set the right camshaft, turn the engine over in the direction of rotation through 45 degrees or until the No. 1 crank is ten degrees past the right dead center. With the crankshaft in this position turn the camshaft over in a clockwise direction until the No. 1 exhaust valve is just closed and the inlet valve is just about to open. Locate the gear in the same manner as in setting the left camshaft.

Before mounting the right distributor driving flange, turn the crankshaft back through 45 degrees or to its original position and set the distributor drive flange so that the marked notch comes in line with the marked tooth on the drive pinion, or in other words, in line with the center line of the right cylinders. Now tighten up the camshaft gear bolts and cotter pin as before.

Timing Ignition.—Set the two distributor assemblies in place, being careful to get them on the proper housings Right and Left. These distributors are marked R and L on the outside surface of the spark control arms. They should be fastened temporarily by means of two bolts No. 116 each in such a position that the notch on the distributor base flange coincides with the notch on the camshaft housing flange. If it has been found necessary to replace either the camshaft housing or the distributor head, and

the new parts do not carry these identifying notches, the distributor should be so set that with the spark retarded the center line of the cylinders will be midway between 1 L and 6 R terminals.

1. Set the engine on the firing point, spark retarded, No. 1 L cylinder, in other words, the neutral point of No. 6 L. Reference to the timing diagram shows that this is ten degrees past the top dead center.

Synchronizing Breakers.—It is assumed that the contact separation or gap was properly adjusted (.010-inch to .013-inch) at the time the distributor heads were overhauled. The synchronization of the two main breakers on each distributor may be checked by inserting a card between the contact points of one breaker and noting the timing of the other. Then repeating the procedure on the other breaker. If the two main breakers on a distributor do not open within one and one-half degrees of each other, they may be synchronized in the following manner: Remove the cotter pin "P" and loosen nut "Q" which will allow the complete breaker arm bracket assembly to be shifted around the stud "AS" as a center. This will vary the setting of the contact separation or gap which should be brought back to the proper limits and the breaker arms tried again for time of opening.

By shifting the breaker arm bracket assembly and each time, bringing the opening of the contact points back to the proper limits, a point can be reached where the two parallel breaker arms will open at exactly the same time, which is the proper setting for these arms. The auxiliary breaker arm must open before the parallel breaker arms when the cam is being rotated in a clockwise direction. If this breaker arm is not adjusted so that it will open in the above manner, it will not prevent the back-firing of the engine when being rocked on compression preparatory to starting. After the distributor breaker arm bracket assembly has been shifted to such a position so as to give the proper opening of the parallel breaker arms, nut "Q" should be screwed down tight and locked in place by cotter pin "P." (See Fig. 442.) The contact screws should be adjusted to line up and meet squarely with the contact points in the breaker arms when the contact points are closed. This can be accomplished by slightly bending and twisting the brackets which hold the contact screws.

2. Swing the timing lever on the distributor to the full retarded position or as far in a clockwise direction as is possible.

3. Loosen the bolts No. 116 sufficiently that the distributor base flange can be rotated on the slotted holes.

4. Connect battery and electric light across the distributor terminals as shown in Fig. 463 B and rotate the distributor base flange in a counter clockwise direction until the light just goes out. Tighten the bolts with the distributor in this position and complete the installation of the bolts No. 116.

5. Without changing the position of the crankshaft install and set the right hand distributor in a similar manner.

6. The accuracy of the timing should now be checked up by rotating the crankshaft backward fifteen or twenty degrees; then forward very slowly, meanwhile watching the electric lights. They should both go out at the same time within a limit of one and one-half degrees on the timing disc. If the pocket flashlight is used instead of the two electric lights and

battery, each distributor head will have to be checked separately and the time of the break noted according to the timing disc.

7. Install the cross reach No 8,333 and adjust it so that both distributor heads will be fully retarded. Check the synchronization of the two distributor heads with spark lever in advanced position also.

8. Install the high-tension cable tube and cable assembly, fastening it by means of the screws No. 201 to the intake headers.

9. Wire the heads of all these screws so that they will not loosen up. **Caution—Care should be exercised in placing the distributor head assembly on the distributor to keep from breaking the rotor brush. It can best be done by putting the distributor head assembly over the two studs as shown in Fig 463 C and slightly rocking it back and forth with the rotor in the**

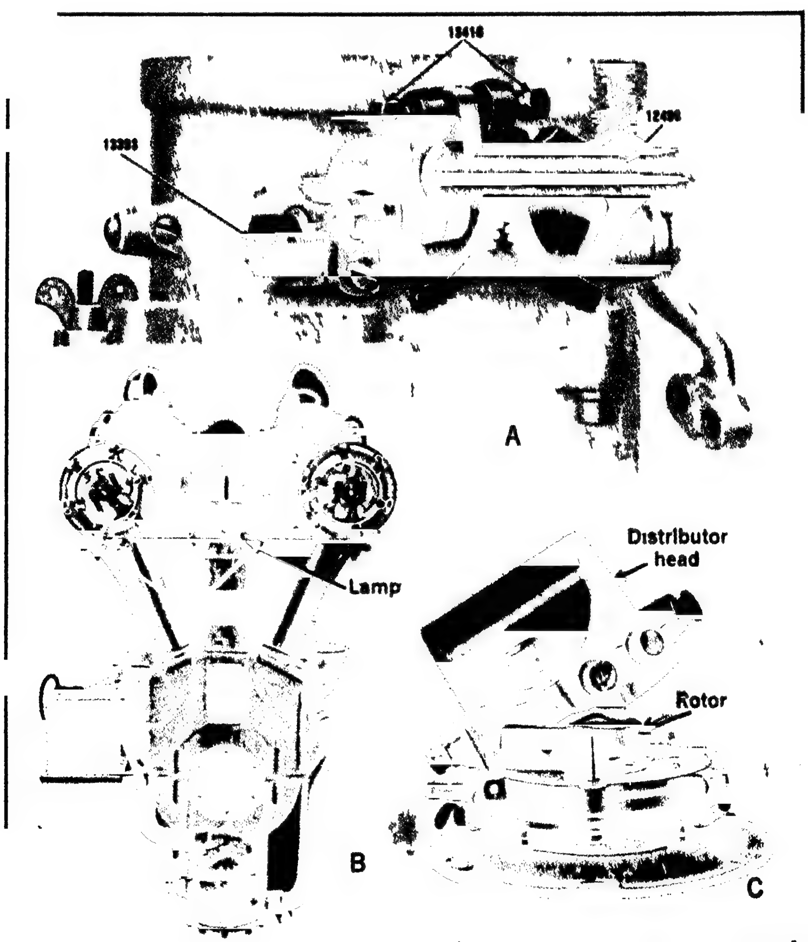


Fig. 463.—A—Method of Synchronizing Throttles. B—Use of Lamps to Check Opening of Breaker Points. C—Method of Replacing Distributor Head on the Engine Base.

right angle position to the center line of the two studs. This will gradually work the brush back into the rotor and allow the distributor head to slip down into place.

Remove wooden plugs from the sparkplug holes and install sparkplugs. Connect the high tension wires to them and to the distributors, being careful that each plug is connected to the proper terminal. Remember that plugs on the side of the cylinders toward the propeller are connected to the left hand distributor and those on the side of the cylinder toward the gear end of the engine are connected to the right hand distributor.

Install the water pump assembly, being careful to see that the gasket No. 8,345 is in place. Try the mesh of the water pump gear with the water pump bevel drive gear. This should be .005-inch minimum and .010-inch maximum. If these gears mesh too tightly, shims No. 8,510 may be inserted between the bearing retainer No. 8,345 and the gasket in sufficient number to produce the proper back lash.

Water Inlet Manifolds.—Install the water inlet manifolds No. 12,123, replacing the hose connections and clamps wherever the original ones were found to be defective. Put on the inlet manifold extensions No. 12,096 and No. 12,097 with their hose connections. In tightening up hose clamps be careful that the clamp is bent so as to fit snugly around the hose and bear against it at all points. Do not draw up the clamps so tightly that the clamp cuts into the rubber. It is advisable before installing the engine in the plane to tape these hose connections with a good rubber friction tape and shellac them, putting clamps on over the shellacked tape. The shellacked tape will protect the hose from oil and gasoline.

Oil Pump Assembly.—Install the oil pump assembly. See that the gasket No. 8,348 is in good condition and is so placed that the gasket does not overlap the oil passages either in the oil pump housing or crankcase. Install the nuts No. 101 and tighten them up evenly and cotter pin them.

Crankcase Breathers.—Install the crankcase breather. See that the gasket No. 8172 is in place and fasten with nuts No. 101. Install the crankcase oil fillers No. 8,156 with gaskets No. 8,158 and fasten in place with nuts No. 101. Install crankcase sump cover No. 8,129. Be sure that the lock No. 8,130 is in its proper place. The whole engine should now be given a careful final inspection. Make sure that all bolts have been properly drawn up and locked either by means of cotter pins or lock wires.

Testing.—Every engine after it has been overhauled should be tested before installing it in a plane. For this testing work the following equipment should be provided:

1. A properly designed test stand. This should be so arranged as to give at least twelve inches clearance between the ground and the tip of the propeller blade when the latter is in a vertical position.

2. A cooling system adequate to keep the engine at the proper temperature. This cooling system should be so arranged that thermometers may be inserted in water passages near the pump inlet and near the water outlet. The temperature of the water at the pump inlet should be about 150 degrees and at the engine outlet about 185 degrees.

SIGNAL CORPS, U. S. ARMY—LOG OF ENGINE TEST

MFG BY __________ __________ AIRCRAFT ENGINE. ENGINE NO __________

TEST MADE AT __________ DATE __________ TEST NO __________

PROPELLER TYPE _____ NO _____ DIA _____ PITCH _____ FUEL _____ B T U PER LB _____ SP GRAV _____ AT _____ °F

WARMING UP HRS _____ R P M _____ OIL _____ VIS _____ SP GRAV _____ AT _____ °F

		PRESSURES		TEMPERATURES			POWER				CONSUMPTION—LBS					REMARKS
					COOLING WATER			BRAKE LOAD								
READING NO	TIME	BAROMETER IN HG	OIL LBS	OUT AIR	IN	OUT	R P M	INDICATED	CORRECTED	B H P (COR)	FUEL PER HR	FUEL PER HP HR	OIL PER HR	OIL PER HP HR	THER EFF RE B H P	WEATHER _____ WIND _____ HUMIDITY _____
AVERAGE																

AV BRAKE M E P _____

AV OIL TEMP IN _____ OUT _____

ACCELERATION _____

MIN R P M _____ H P _____

MAX R P M _____ H P _____

VIBRATIONS _____

COMPRESSION _____

IN CHARGE __________ OBSERVER __________

Fig. 464.—Log of Engine Test Used by Aviation Section, Signal Corps, U. S. Army which May be Used as a Model in Preparing Similar Blanks for Recording Engine Trials, Such Changes Being Made as Necessary. The S.A.E. Has a Good Standard Test Form, Also.

3. An oil reservoir with suitable connections to the engine so arranged that a thermometer can be inserted in the line between the oil pump outlet and the reservoir. This thermometer should preferably be fitted in a temporary oil sump cover (propeller end) so that it will extend up into the oil connecting in this sump.

4. A control board should be arranged with oil pressure gauge, tachometer switches, ammeter and voltage regulator.

In preparing the engine for a test the instructions given under the head of "Preparing an Engine for Service" should be followed out.

After the engine has been started and while it is warming up, careful attention should be paid to the operation of the oiling system. If the pressure gauge does not show the proper pressure or any bearing seems to be warming up excessively the engine should be stopped immediately and the trouble investigated. The engine should be run at a speed of about 1,200 r.p.m. for at least one hour. If the engine has been completely overhauled and new bearings have been fitted, it is advisable to connect the oil pump inlet and outlet in such a manner that the oil will be bypassed. Then fill the crankcase with three gallons of oil. This will be a sufficient amount to permit the connecting rods to dip into the oil and will insure excessive lubrication during this period. After an hour's run the bypass connections may be taken off, oil drained out of the crankcase, and the pump connected to the oil reservoir. The speed of the engine may now be gradually increased up to 1,400 r.p.m. and held at this speed about one-half an hour. The throttle may then be opened wide for a period of three to five minutes, under which conditions the speed should increase to about 1,600 r.p.m. If there is no decrease in speed during this period, the engine may be considered satisfactory.

The operator during the progress of the test should watch the water and oil temperature and the speed in r.p.m. and make note of these points at fifteen-minute intervals. He should also watch for water and oil leaks around the engine. Any appreciable dropping off in speed with no change of the throttle position is an indication of excessive friction being developed on account of bearings being too tightly fitted or the improper functioning of the engine, either due to ignition or carburetion. The engine should be shut down and the trouble investigated.

SUMMARY OF CLEARANCES

Taken from Standard Drawings of Liberty 12 Engine

	Minimum	Maximum	Desired
Crankshaft			
Diametrical Clearance	.0025	.00325	
End Play	.0575	.0775	
Connecting Rods			
Forked End			
Diametrical Clearance	[illegible]	[illegible]	
End Play	.008	.020	
Plain End			
Diametrical Clearance	.005	.0065	
End Play	.004	.008	

SUMMARY OF CLEARANCES—*Continued*

Piston Pin			
Fit in rod	.00025	.00125	Select for .001 Clearance
Fit in piston	.00025 loose	.00075 tight	Select for light drive fit
Piston Rings			
Fit in grooves	.00125	.003	Top .003 Mid. and Bot. .002
Gap	.021	.041	.030
Piston			
Fit in cylinder	.018	.022	Select for .020 Clearance
Camshaft			
Diametrical Clearance	.001	.003	
End Play	.000	.004	Min. .002
Camshaft Upper Drive Shaft			
Diametrical Clearance			
Large bushing	.0005	.0025	Min. .0015
Small bushing	.0005	.0025	Min. .0015
End Play	002	.008	Min. .004
Rocker Levers			
Diametrical Clearance	.00025	.00175	Min. .001
End Play	.005	.010	.0075
Valves			
Fit of stems in guides			
Diametrical Clearance			
Exhaust valve	.004	0065	.005
Inlet valve	.002	.0045	.003
Water Pump Shaft			
Diametrical Clearance	.0015	.0035	Min. .0025
End Play	.006	.010	.010
Water Pump Bevel Driver			
Diametrical Clearance	001	0025	
End Play	005	.008	
Oil Pump			
Fit of gears in housing			
Diametrical Clearance	.001	.005	Select for .004 Clearance
End Play	.002	.007	Select for .003 Clearance
Tappet Gap			
Exhaust valve	.019	.021	
Inlet valve	.013	.016	
Breaker Gap	.010	.013	
Sparkplug Gap	.015	.017	.015
Regulator			
Contact gap	.005	.007	
Height of pin	.043	.045	

Caution—If the engine being tested is designed for use in high altitudes, it is equipped with crowned high compression pistons. It should not be run on the stand with throttle more than one-half to two-thirds open. The crowned pistons have a clearance volume so worked out that the compression would be excessive at low altitude and running it with wide open throttle on the ground would ultimately result in a breakdown. Engines fitted with flat top low compression pistons may be run on the stand with wide open throttle. During the test on the stand the altitude adjustment should be fastened in a closed position.

The log of the engine test should be kept carefully on suitable forms provided for that purpose. A sample form used in connection with a suitable dynamometer installation is shown at Fig. 464 and will be found of value by any one testing engines of any kind. Standard test forms may be obtained from the Society of Automotive Engineers, Inc.

Vee Type Inverted Liberty Air-Cooled Engine.—The Vee type engine is particularly well suited to air-cooling, since with a minimum amount of cowling, a rapid air flow is directed on all surfaces of the cylinder barrels and heads. In fact, the engine itself might be said to form its own cowling. A single sheet of aluminum arched across the open part of the Vee, together with a back plate of aluminum, forms a deep box, which receives air from the slipstream of the propeller, and distributes it between the cylinders. As the air intake area of the cowling is some twenty per cent greater than the area of the outlet passages, sufficient pressure is built up to force the air to flow out at high velocity between the barrels, heads, and valve ports, which insures adequate cooling of these parts. The cooling air finally passes out of the ports in the side of the airplane cowling and away from the pilot.

The favorable consideration of air cooling, which forms perhaps the most striking feature of present day development in the field of aircraft engines, was clearly realized by the Engineering Division of the Army Air Corps at McCook Field, when more than five years ago, it conceived the plan of air cooling the Liberty engine. Preliminary layouts and calculations were made, and a proposal for bids was issued, calling for the design and construction of an experimental engine. The Allison Engineering Company was selected to carry out this project. The first engine, designed and built for upright operation and shown at Fig. 464 A, was so successful that the Air Corps was encouraged to continue in its development. It was decided however, that because of better visibility the engine should be designed to run in the inverted position. A further advantage of the inverted type is that even with short air-cooled exhaust stacks, which minimize fire hazard, the gases and noise of the exhaust are directed away from the cockpit. An end sectional view of this engine is shown at Fig. 464 B. This drawing outlines the cylinder construction very clearly.

The addition of a rotary induction system permits the use of straight intake manifolds and single carburetor. The camshaft housings are made quite large and deep to form oil sumps into which the valve stems and springs dip. Not only does this insure thorough lubrication of the valve mechanism, but as the exhaust valve stems, of large diameter, are made hollow and

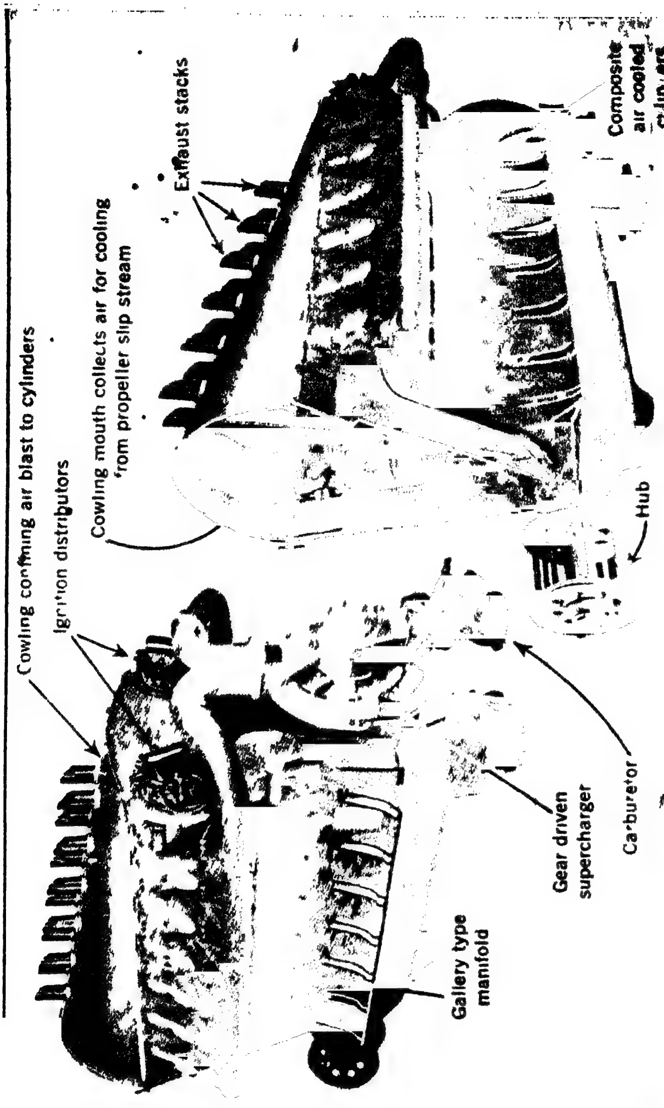


Fig. 464A.—Views of Liberty Air-Cooled Aviation Engine in Which Flanged Cylinders Replace the Water Jacketed Cylinders Originally Used.

partially filled with a fusible salt mixture, the agitation of which carries heat from the valve head to the stem end where it is transferred to the oil, it thereby forms an essential part of the cooling system. The oil is scavenged from both ends of the housings by double gear pumps located on the aft end of each housing, from which it is discharged into the timing gear housing at the aft end of the crankcase.

The cylinder barrels project approximately two inches inside the crankcase, the space around them forming an adequate sump which collects the oil thrown off the crankshaft bearings, that scraped off the cylinder walls, and the discharge from the camshaft scavenging pumps. A double scavenging pump, on top of the crankcase pan, collects the oil from both ends of

this sump and forces it through the jackets on the carburetor and carburetor elbow, and thence to the supply tank. Built integral with the scavenging pump is the pressure pump, which maintains a pressure of 100 pounds per square inch at the crankshaft bearings. A somewhat lower pressure is used on the camshaft bearings.

Materials used in the various parts of the cylinder assemblies were selected as best suited for the particular work each part is subjected to. The barrels with their integral cooling fins and hold-down flanges are machined from tempered steel forgings. On the head end is machined a twelve pitch thread, one and one-half inches long for attaching to the cylinder heads. The valve seats and sparkplug inserts are of tempered

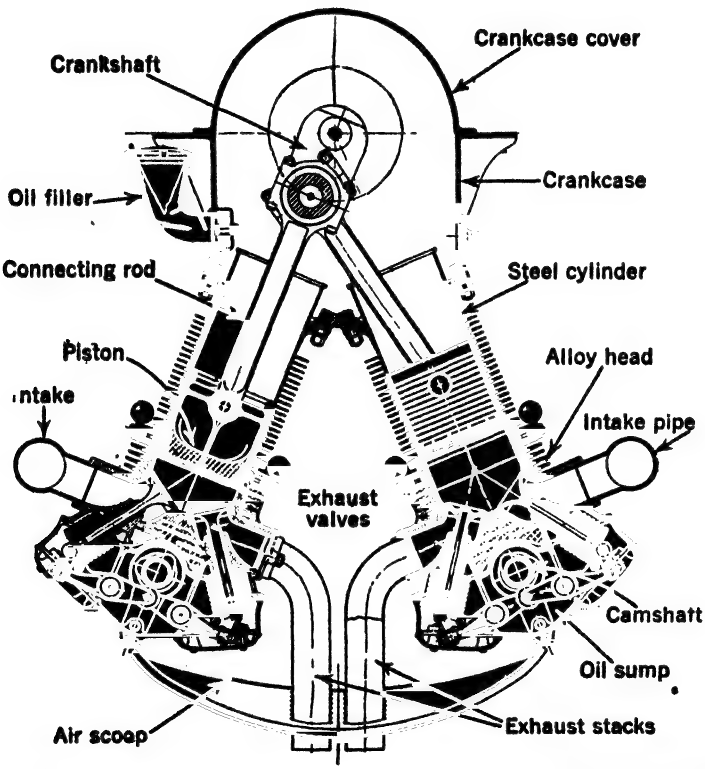


Fig. 464B.—Sectional View of Inverted Liberty Air-Cooled Vee Engine Showing Parts.

aluminum bronze. The cylinder heads, with their integral cooling fins, are cast of aluminum (Y Alloy), and are heat treated before machining. They are machined to receive the threaded end of the cylinder barrel and the valve seat inserts, drilled and reamed for the valve stem guides, and are tapped for fitting the sparkplug inserts; the bores and threads in the heads being somewhat smaller than the outer diameter and threads of the parts

fitted into them. To assemble the various parts it is necessary to heat the cylinder head to a temperature of approximately 650 degrees Fahrenheit, the other parts being at room temperature.

When the cylinder head is cooled, the component parts of the assembly are held together under considerable tension. This method of assembly makes sure of good heat flow through the joints and although the cylinder assembly, in operation, will be worked at a fairly high temperature, the difference in temperature of the various parts will never be great enough to allow their loosening. After the cylinder assembly has been made up as above it is subjected to a hydrostatic test of 650 pounds, after which the valve seats are finished and the cylinder bore ground to size.

SPECIFICATIONS:

Name	Allison
Model	V-1410 and VG-1410
Arrangement	Inverted V-45 deg.
Cooling Medium	Air
Bore	4⅝ in.
Stroke	7 in.
Total Piston Displacement	1410 cu. in.
Compression Ratio	5.3 to 1
Max. Propeller r.p.m.	V-1410· 1900; VG-1410: 1140
Horsepower	410 at 1880; 430 at 1900
B.m.e.p.	128 lb./sq. in. at 1800 r.p.m.
Max. Gas Consumption	55 lb. per hp. per hr.
Max. Oil Consumption	.03 lb. per hp. per hr.
Carburetor	Stromberg, Type NA-S8E
Ignition	Delco-Remy 12 Volt Battery
Weight	V-1410: 1025 lb.; VG-1410: 1160 lb.
These weights include generator, propeller hub, exhaust stacks and cowling, but not the starter.	
Length	V-1410: 78½; VG-1410: 85⅛ in.
Height over cowling	Front: 46¾ in.; Rear 44$\frac{7}{16}$ in.
Width	34½ in.
Height above engine bed	14¾ in.
Center to center engine bearers	17 in.

The centrifugal (high speed fan) type of rotary induction is used on the air-cooled Liberty. By its use a single carburetor can be employed to supply all twelve cylinders. Furthermore, the high speed of the impeller effects such thorough mixing and atomization that all cylinders receive a uniform mixture, although the simplest form of intake manifolds are used. A valuable characteristic of the centrifugal fan is that both capacity and pressure increases more rapidly than the speed, and so the amount of charging increases with the speed, counteracting the natural decrease of volumetric efficiency which would result if the intake depended on suction only. Thus at 1,800 r.p.m. on wide open throttle the pressure in the manifold is approximately one and three-quarter inches of mercury above atmosphere and at 1,900 r.p.m. two and one-quarter inches. The entire rotary induction assembly is built up as a unit, and mounted on the aft end of the Liberty crankcase, being bolted to the flanges ordinarily used for mounting the starter and water pump. The ten to one step up gearing is connected to the crankshaft through a constant torque plate clutch which prevents breakage of teeth due to severe loading caused by rapid acceleration and

torsional deflections of the crankshaft.

The fan consists of a hub with twelve straight radial blades, with an outer diameter of six and one-half inches machined from a solid forged disc of duralumin. The mixture is drawn from the carburetor to the center of the impeller, but instead of flowing axially into the fan, which would cause turbulence due to the blades meeting the gases at right angles, it is deflected in the direction of the impeller discharge by stationary curved blades in the entrance to the impeller chamber. The mixture issuing at high velocity from the impeller, is delivered through a narrow annular passage to the diffusion chamber, and thence through a Y fitting to the intake manifolds connecting the right and left banks of cylinders. On the back face of the rotary induction housing is a standard starter mounting flange —the drive from the starter to the crankshaft being through constant mesh idler gears.

The propeller reduction gearing used in the Model VG-1,410 is the Allison Epicyclic type and is fitted with a spring coupling which effectively prevents breakage and excessive wear of the gear teeth. The reduction ratio is five to three. Propeller rotation, same as crankshaft. The use of this gear requires a special crankcase and crankshaft.

QUESTIONS FOR REVIEW

1. Outline necessary precautions in removing Liberty and similar engines from snipping crates.
2. Describe steps in process of preparing Liberty engine for service.
3. What parts of the ignition system need inspection?
4. What is done to start Liberty engine?
5. Outline some of the causes of trouble in Liberty engines.
6. How often should Liberty engines be overhauled?
7. What is the electrical equipment of the Liberty engine and why is battery ignition used?
8. Outline points in camshaft housing units needing inspection.
9. Describe construction of Liberty engine oil and water pumps.
10. What process of timing is used to insure synchronization of ignition and why is it necessary?

END OF VOLUME ONE

Complete Installation, Trouble Shooting and Repair Instructions for All Leading Aircraft Engines of Recent Construction Given in Volume Two. See Table of Contents for Summary of Data.

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